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(Continued on page 34)

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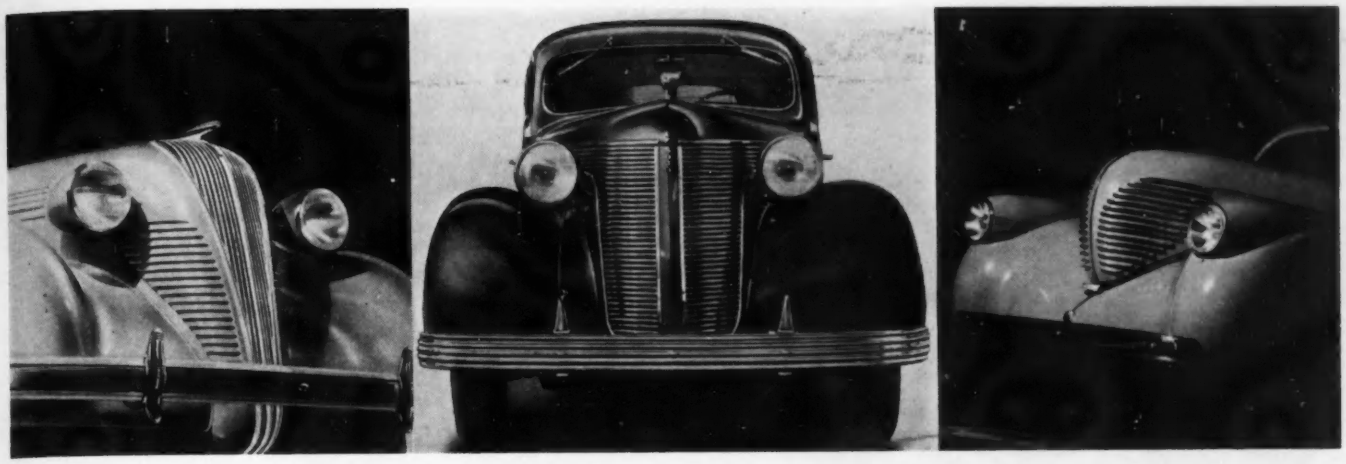


The New Models March On! Here Are Terraplane, DeSoto and Willys

Trends in 1937 Car Design

By Austin M. Wolf
Consulting Engineer

ALTHOUGH no radical changes are to be found in the 1937 cars, the number of detailed improvements is probably greater than in any other year and, collectively, they contribute to a new standard of design and performance. Bodies, sheet-metal work, and grilles have reached a new height of beauty but are clearly in the transient stage toward a more complete yet practical streamlined design. Many safety measures have been incorporated.

General

Newly introduced models are the revised V-8 engined LaSalle with 124-in. wheelbase and the Packard-6 with a 237 cu. in. engine and 115-in. wheelbase. The LaSalle is equipped with the former Cadillac "60" 322 cu. in. engine, superseding the 248 cu. in. straight 8. In its large models, Packard has dropped the previous 8 and confined itself to the super-8 and 12. The Chevrolet Master and Master Deluxe have the same basic chassis except for individual front-wheel suspension in the latter. Pontiac has increased wheelbases by 5 in., 117 in. for the 6 and 122 in. for the 8. Master models have been discontinued with the Deluxe as the sole production. The Willys, dressed ultra-modern fashion, has 56-in. tread and, although retaining 100-in. wheelbase, has an 11½ in. greater overall length than the previous model. Hudson, Terraplane, and Oldsmobile-6 have added 2 in. to their wheelbases; Oldsmobile-8 and Lafayette, 3 in.; Buick "40" and "60," 4 in.; while Chevrolet has reduced it by ¾ in. with a weight reduction of from 3255 lb. to 3115 lb.

Engines

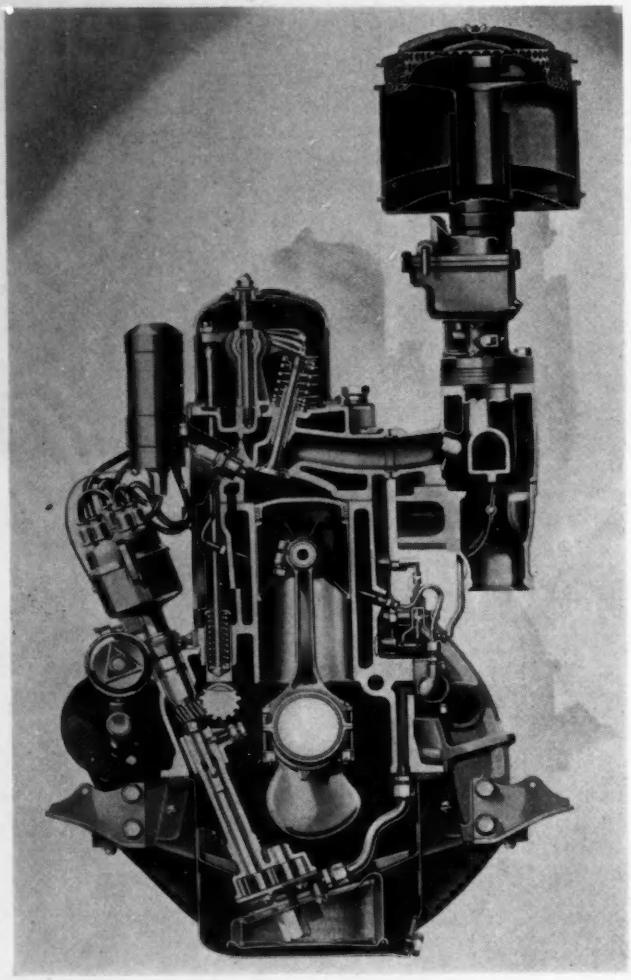
The redesigned Chevrolet engine has four-bearing crankshafts and camshafts, slipper-type lighter-weight pistons, 216.5 cu. in. displacement (3½ in. x 3¾ in.), 6.25:1 compression ratio, and develops 85 hp. at 3200 r.p.m. Maximum torque is 170 ft-lb. between 900 and 2000 r.p.m.

The Packard-6, 3$\frac{7}{16}$ in. x 4¼ in., 6.3:1 compression ratio, develops 100 hp. at 3600 r.p.m., has a cast-iron head, and weighs 695 lb. with clutch and transmission. The Packard Super-8 engine is several hundred pounds lighter than the former 8, weighing 976 lb. with clutch and transmission, and develops 135 hp. at 3200 r.p.m.

All the Cadillac V-8s are powered by the 346 cu. in. engine, comprising Models "60," "65," "70," and "75." The Buick "40" has a ¼ in. greater stroke, a compression ratio increase of from 5.65 to 5.70:1, raising the displacement to 248 cu. in. and the hp. from 93 to 100. The engine of the larger model remains the same size, but the compression ratio has been stepped up from 5.45 to 5.75:1 and the hp., from 120 to 130.

Retaining the same cylinder size and compression ratio, Terraplane has increased its output from 88 hp. at 3800 r.p.m. to 96 hp. at 3900 r.p.m. with a cast-iron head. With an

[This paper presented at the Regional Transportation and Maintenance Meeting, Newark, N. J., sponsored by the Metropolitan, Philadelphia and Southern New England Sections of the Society.]



Chevrolet Engine with Compact Combustion-Chamber, Four-Bearing Crankshaft, Slipper Pistons and Gear Oil Pump

aluminum head and 7:1 ratio it develops 101 hp. at 4000 r.p.m. Similarly, the Hudson-8 has remained unchanged but develops 122 hp. at 4200 r.p.m. as compared with 113 hp. at 3800 r.p.m. last year.

The Oldsmobile-6 displacement has been increased from 213 to 230 cu. in. and the hp. raised from 90 to 95 at 3500 r.p.m., whereas the 8's displacement has been raised from 240 to 257 cu. in. and the output from 100 to 110 hp. at 3600 r.p.m. Both engines have been developed in order to obtain interchangeability among as many parts as possible, comprising the timing chain, sprockets, valves, valve guides, springs and tappets, connecting rod, water pump, fuel pump, clutch housing, oil-pump drive, generator, oil filler, and spark-plugs.

The Pontiac-6 has changed from 3⅜ in. x 3⅞ in. to 3 7/16 in. x 4 in., raising the displacement from 208 to 223 cu. in., the hp. output from 81 to 85, and maximum torque from 150 to 161 ft-lb. A ¼ in. greater stroke on the 8, from 3½ in. to 3¾ in., has raised the displacement from 232 to 249 cu. in., the hp. from 87 to 100, and maximum torque 161 to 173 ft-lb. The compression ratio, 6.2:1, was unchanged. By increasing the bore ⅛ in., the Lafayette now has the same cylinder size as the Nash-6, 3⅜ in. x 4⅜ in. At 3400 r.p.m., the former develops 90 hp. (83 hp. at 3200 r.p.m. last year) and the latter 95 hp., a credit of 5 hp. due partially to overhead valves.

The Studebaker, Plymouth, and Dodge engines remain the same, but it is interesting to note that the Chrysler Royal and DeSoto engine, 3⅜ in. x 4¼ in., has ¼ in. less stroke than last year's 6, reducing the displacement from 241.5 to 228.1 cu. in. Compression ratio has been stepped up from 6.0:1 to 6.5:1 and the hp. output remains at 93 but at 3600 r.p.m. instead of 3400 r.p.m. last year.

Where engine output has been increased, a faster axle ratio is used to decrease the engine r.p.m. and the piston travel per mile. With weight reductions that have been made in numerous instances, fuel economy has been bettered at the same time.

Cylinder and Head

Besides adding a full-length water jacket, Oldsmobile has tackled cylinder distortion by placing a rib around the barrels about 1¼ in. down from the top deck, relocating and altering the shape of the water holes, and by the addition of intermittent grooving around the barrels to distribute the loading stresses from the bolt boss uniformly to the barrel. The Buick cylinder-head bolts are blind-tapped into the block. This is also the case with the rocker-shaft bracket bolts in the head except for one hole which is sealed with a special tapered stud. Cadillac is using a lighter cylinder-head to reduce cylinder-block distortion as affecting the bore and valve seats and to reduce any tendency toward gasket leaks, the block itself being made stiffer.

The Federal-Mogul Thermo-Flow copper-alloy head is now in production for the Ford V-8 engine. Briggs Mfg. Co. has developed a composite cylinder-head in which the combustion-chamber section and face is cast iron with a pressed-steel water-jacket cover automatically seam-welded to a steel flange cast therein.

Valve Gear

The Wilcox-Rich self-adjusting hydraulic tappet is used by Cadillac, LaSalle, Pierce-Arrow, and Lincoln-12. In the Domark engine the hydraulic-tappet piston has restricted travel to prevent undue opening of a stuck valve. In order to keep up the idling oil pressure, the holes in the camshaft bearings have been reduced to raise the pressure to approximately 5 lb. per sq. in. higher than formerly. The reservoirs that were provided in the Cadillac valve-tappet brackets have been eliminated, the oil passing directly to each tappet body through a longitudinal passage drilled through the guide casting after flowing through a Handy oil filter. The filter prevents dirt from entering the tappets and also serves as a trap for air which might otherwise make them noisy. An additional safeguard against air-trapping is the use of a "jiggle pin" in a cap in each casting with a clearance of 0.006 to 0.010 in. A continuous small leakage of oil ensues and plugging is prevented by the pin moving up and down with every change of oil pressure.

The Wilcox-Rich self-locking tappet adjusting screw eliminates the necessity for the conventional locknut and provides greater adjustment accuracy. It is used on the Chrysler-6 and -8 and DeSoto engines and consists of a threaded idler which is keyed so that it can be slid to a reduced portion of the screw body. The idler thread is a continuation of the thread on the screw proper except that it is mismatched and is retained in position by two washer-type springs riveted to the lower end of the shank. With the general introduction of cast-iron camshafts, the steel mushroom tappet has come into favor since it combines excellent wearing qualities with extreme light weight. Oldsmobile is using such a camshaft with a rounded-

cam-tip contour, thus minimizing any tendency toward pitting or excessive wear of either the tappet or camshaft. The lighter-weight tappet permits a substantial decrease in valve-spring load. These tappets operate in bosses cast in the block, whereas last year's 8 had a bolted-on cluster. Spring surge is minimized by a cylindrical damper extending down within the top few coils. The Hudson Terraplane tappet face has been changed from 3-in. to 2-in. radius, reducing the lift velocity slightly and making the action smoother.

Oldsmobile has adopted Silchrome XCR for the exhaust valves, whereas Willys uses Silchrome No. 1 with an inserted seat. Buick's streamlined inlet valve has been a factor in increased output. Pontiac has adopted the split-cone type valve retainer.

Chevrolet continues the rocker-arm shaft oil-feed tube through the water jacket. Valve-lash control on the large Buick engine, in which the time and lash characteristics are practically the same as on the 1936 series, is obtained by the expansion or contraction of a die-cast aluminum rocker-arm bracket.

Link-Belt ½-in. pitch chains of 1-in. width are used by Buick "40," Graham, and Oldsmobile; whereas 1¼-in. width is used on the large Buick engine and Willys, all non-adjustable; ⅜-in. pitch is used by Duesenberg and Stutz, each with automatic adjustment. Morse stiff-back chains are used on all Packards including the new 6 which interchanges with the "120," being ⅜-in. pitch and 1¼-in. width.

Piston and Rings

The Chevrolet light-weight pistons have a beveled roof, permitting a more compact combustion-chamber and eliminating the pocket over the side opposite the plug. A reinforcing rib follows each slipper edge. The possibility of distortion in the Oldsmobile pistons has been minimized by slightly increasing the thickness of the skirt wall, revising the skirt reinforcing area and by the use of double-top deck ribs. Besides improved ring operation, it has permitted the clearance between the pistons and the cylinder bore lengthwise of the engine to be increased approximately 0.003 in. on the side over 1936. Starting friction is decreased. To balance the effect of the locking screw in one piston boss, additional metal is cast on the opposite boss. The balancing rib to maintain the overall weight of the piston is notched to allow uniform expansion of the skirt regardless of the amount of metal removed for balancing.

The Bohn Autothermic tin-plated pistons are used in the Packard-6, the entire Graham line, and the Dodge truck. Bohnalite "L," a 12 to 14 per cent silicon alloy, is used and tin-plated by the immersion process. All Chrysler products have a U-slot in place of the former T. Campbell, Wyant & Cannon has a considerable number of molybdenum-silicon steel pistons in the field with the virtue of added strength and a lightness almost equal to aluminum. Cadillac pistons are graded into one of thirty sizes and a similar division is made of all cylinder bores. This grading permits a selective fit on assembly with a clearance variation not greater than 0.00007 in.

The Perfect Circle X-90 oil and compression rings utilize a series of independent double-leaf spring units equally spaced on a carrier band the gap of which is located opposite the ring gap. Uniformly maintained pressure is provided, and this arrangement is standard equipment in Packard and Graham. Packard claims a 200 per cent improvement in oil economy at 75 m.p.h., and less oil consumption was shown at 35,000 miles than at 10,000 miles previously.

The Sealed Power compression ring has a slight taper machined on the peripheral face, this being 0.0005 in. in the case of Buick. It allows the rings to seat-in quicker and bring about a better seal. The mirror-hone finish of cylinder walls has created an additional problem requiring accuracy in design and workmanship. Plain rings often require 800 to 1500 miles to seat properly. The tapered face permits the compression ring to function as an oil scraper during the breaking-in period. The word "top" is stamped on the proper side to insure correct installation. Compression rings 3/32 in. wide are used in increasing number rather than the ⅛ in. width due to greater flexibility permitting conformation to the irregularities of the cylinder wall. Oil rings 3/16 in. wide are used generally, replacing the former 5/32 in. width and providing a sufficient amount of ring tension to insure satisfactory oil consumption at all operating speeds, and also permitting greater circulation of oil through the larger oil-vent slots which are less prone to clogging with carbon or sludge accumulation.

Chevrolet is still provided with three rings, whereas Willys uses four: three compression and one oil control. The Willys wrist-pins are full-floating, being free to rotate in a bronze-bushed connecting rod upper-end and in the bosses of the cast-iron piston. The Domark engine, with two ⅛-in. compression rings and one 3/16-in. oil ring, locates the latter ring in the lower part of the piston in the zone of low temperature in view of the saw slot separation from the head. Graham has two compression and two oil rings, all above the pin.

Connecting Rod

With slightly lessened weight, the Cadillac V-8 engines have increased the stiffness of both the rod I-beam and the bearing backing, increasing the factor of safety against rod breakage and increasing bearing life. The Oldsmobile-6 and -8 rods have been redesigned completely and are interchangeable between the two models except for an extra machining operation on the 8 rod to permit installation through the cylinder barrel. Complete service interchangeability is possible as the 8 rod can be used in both engines. A 6 per cent weight reduction on the 6 and 16 per cent on the 8 over the previous designs includes the removal of the rib on the bottom of the cap, the adoption of the uniform section around the big end without any stress concentration point, and a smaller I-beam section blending with a larger radius into the big end. Graham has made a 4-oz. reduction in its connecting-rod, piston, and ring assemblies, and cadmium-silver bearings are used in the big end. In place of the conventional babbitt liner, Buick is using non-crystallizing babbitt, doubling the rod life under extreme conditions.

Crankshaft

The four-bearing Chevrolet crankshaft has improved operation considerably. To minimize high-speed vibration, the Cadillac V-8 engines use a lighter flywheel and a torsional-vibration damper has been added to the 346 cu. in. engine. Thicker cheeks are used in the Pontiacs and the counterweighting has been increased from 80 to 100 per cent. To avoid roughness both Pontiac and Oldsmobile have thickened the flywheel flange as well as the flywheel section at that point. The Oldsmobile rear oil seal consists of a special braided-asbestos wiper contacting the machined surface of the shaft, retaining oil even if the level is above the seal. The Tocco hardening process of the Ohio Crankshaft Co., used

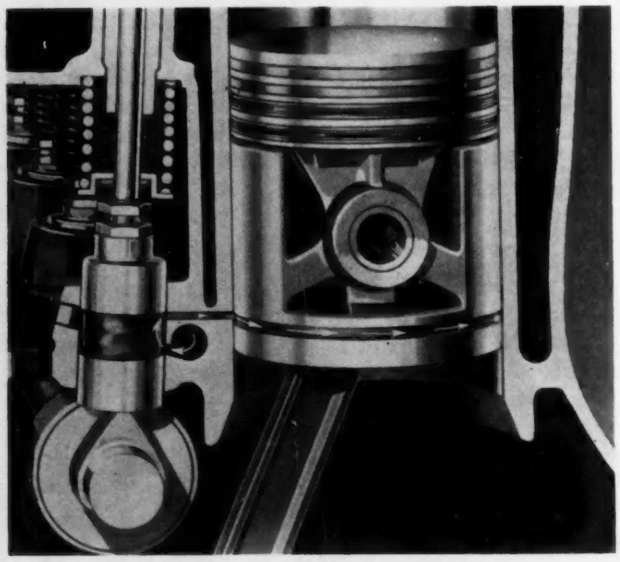
by General Motors, Packard, White, Hercules, Waukesha, and International Harvester, consists of passing a low-voltage, high-amperage current through inductor plugs surrounding the shaft but not actually touching the bearing area. The induced current heats the surface metal and, after correct length of soaking time, the circuit is opened and quenching is obtained by spray from a water jacket built in the inductor plug. The Packard-6 81½-lb. four-bearing crankshaft has six integral counterweights with thrust taken on No. 1 main bearing with an end play of 0.003 to 0.008 in. Main bearings are 2¾ in. and fitted within 0.002 in. clearance. Packard vibration dampers are of the rubber and friction type. The Pontiac harmonic balancer ½-in. pins are elliptical with a 1-in. radius where they contact with the balancer springs. Buick is using a 3/32-in. noncrystallizing-babbitt main-bearing liner in place of the previous 5/32-in. babbitt liner.

Lubrication

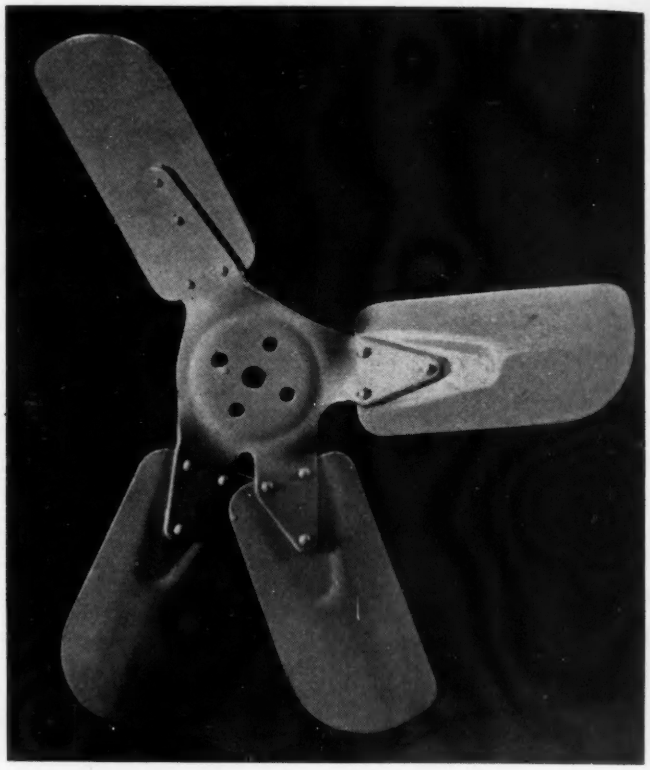
The Graham pistons are lubricated by a metered flow of oil that is bypassed in a circumferential relief in the valve tappet in its topmost position from a supply port to a groove near the bottom of the skirt. A vertical groove controls the length of time that the piston is in communication with the pressure line, and an air-vent hole on the opposite side stimulates the flow of cold oil around to that point quickly. With this system of copious supply and Perfect Circle X-90 rings it has been possible to average 1000 to 1200 miles per qt.

Buick has eliminated the oil filter and instead uses the floating oil-pump screen. Chevrolet has adopted a gear-type pump located immediately above the screen and inverted bell. Oil, gasoline, and water are all serviced from the right side of the car.

Studebaker, in adopting the Fram cleaner, recognizes only summer and winter oil changes if a new cartridge is installed whenever the dipstick shows the oil to be dirty. The Cadillac-16 uses a Handy filter in place of the previous Cuno. The Model N Purolator uses a vegetable product, Multocell, which continues to retain contamination from the oil passing through it to a point where even the oil itself will no longer pass. Packard has added an air cleaner to the crankcase filler and ventilator pipe. Federal-Mogul has introduced a service-station bearing oil-leak detector to check all lubrication lines and bearings.



Graham Metered Piston Lubrication



Hayes Fan Used on Buick "80" and "90"

Cooling System

The Buick spring-loaded bypass has a fixed orifice equal to a ½-in. diameter hole, permitting anti-freeze mixture to circulate freely without loss. The Series "40" and "60" models have had the frontal core area stepped up from 402 to 425 sq. in., while the core depth has been decreased from 2⅛ in. to 2 in. This design is in line with the attempt of Harrison Radiator Co. to standardize on heights, widths, and depths. They have found improved dissipation from the 2-in. depth so that it equals the original 2⅛-in. core dissipation, brought about by changes in the air flow. Although retaining the same frontal area (405 sq. in.) the Oldsmobile models have a top tank capacity of 200 cu. in. as against 83 cu. in. on the previous 6, and 134 cu. in. on the previous 8. The pump impeller has been increased in size, giving a capacity of 55 gal. per min. at 4000 engine r.p.m. The Cadillac core has been decreased from 4 in. to 3½ in. and the pressure-valve cap has a bayonet-type fastener instead of the former threads. The first rotation toward the left vents the cap to the atmosphere to relieve any built-up pressure, and more forceful rotation to the left permits its removal.

The overall length of the Packard Super-8 pump has been increased, permitting the bearings to be moved farther apart to resist better the pull of the single V-belt. The Oldsmobile pump is equipped with a permanently lubricated double-row ball-bearing in which the hardened impeller shaft has grooves for the balls to roll in. The pump seal consists of a flat graphite washer adjacent to the impeller, backed with a Duprene seal and a bronze spring to maintain constant pres-

sure on these parts. Buick now uses a chevron-type automatic packing with a Duprene annular ring.

Pontiac has dropped its pump speed from 4300 to 3030 r.p.m. at 60 m.p.h. but circulates more water (2600 gal. per hr. at 60 m.p.h.), because of the increased impeller diameter of 3¾ in. over 3¼ in. The fan revolves at 30 per cent lower speed and its diameter is increased from 16 in. to 17⅞ in. The fan belt on the Cadillac V-8 engine now drives the fan only, the load being reduced by driving the water pump with the generator belt. Eliminating the triangular drive provides greater pulley contact. The Lincoln-Zephyr fan has alternate short and long blades with oval-shaped ends. The Hayes Industries four-bladed fans on the Buick "80" and "90" are notable for their quietness; blades have a spacing of 113, 76, 42 and 129 deg.

Fuel System

The Stromberg Model AA carburetor, used on the entire Buick line, has a float on either side connected to a single operating lever which, in conjunction with baffles between the mixing-chamber barrels, maintains the proper height of fuel in each nozzle at all times. In providing an annular space around the main discharge jet, the fuel acts as insulation to heat carried through the throttle-valve body. Furthermore, all fuel entering the jet must pass through a series of small holes which will filter out all bubbles that have accumulated. They escape through the domed high-speed bleeder which acts as a vent when the engine is not running. The Packard-6 is equipped with the Chandler-Groves carburetor in which the main jet is formed in a bridge above the venturi. A two-piece float is used. Graham continues with Marvel equipment.

The Super Terraplane is equipped with a dual Carter carburetor, one mixing chamber feeding the first three cylinders and the second, the remaining three. The characteristic cylinder porting is retained and a balance passageway is established between the branches at the back of the inlet manifold. The Hudson-8 now has dual carburetion. The Willys is equipped with the Tillotson 1⅛-in. down-draft variable-venturi carburetor.

The Delco-Remy automatic control, used on all Buicks, is mounted on the back side of the carburetor above the exhaust manifold and operates the choke valve by means of a flexible cable. The unit is sensitive to manifold temperatures, intake-manifold vacuum, carburetor-air-inlet velocities, and throttle opening. A fuel-volatility selector is provided. Cadillac chokes are now fully automatic in retaining the electrical portion of the previous triple-range unit. Dodge, DeSoto, and Chrysler continue with the Sisson automatic choke. The Stromberg choke used by Studebaker is provided with a fuel-volatility adjustment. The Carter and Delco-Remy units provide for automatic de-flooding by releasing the choke mechanism when the throttle is fully open.

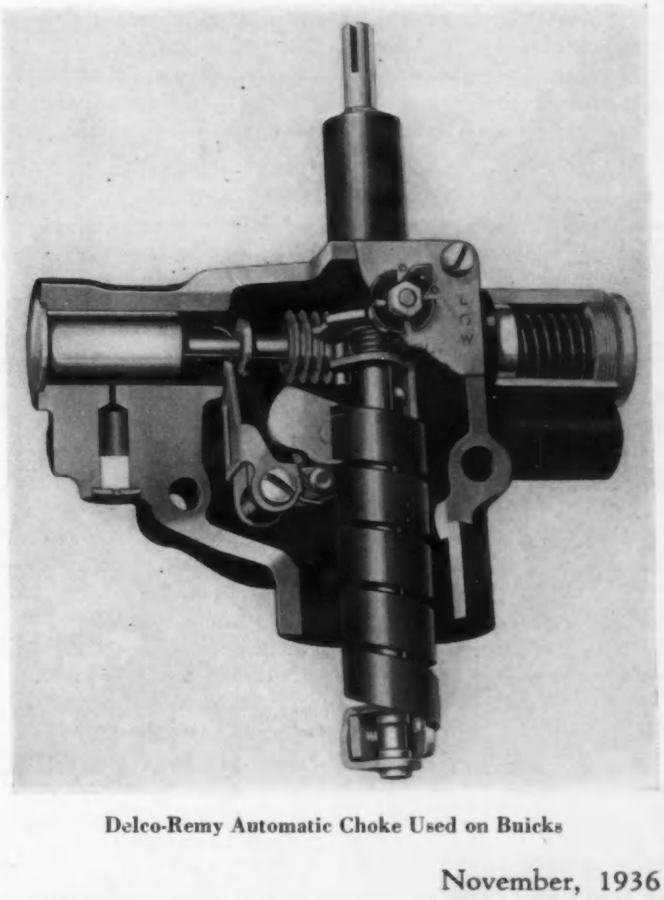
The Oldsmobile-6 manifold has branches of three-fourths round section instead of a flat bottom to provide better vaporization. A smaller hot-spot has been incorporated to decrease residual heat. The 8 manifold has been altered to suit the four-bolt Carter carburetor flange. More uniform distribution and better heat control are the general objectives. Pontiac has accomplished this purpose by changing the position of the steel-riser portion of the manifold, whereas Chevrolet has an entirely new design.

The oil-bath type of air cleaner is very popular. On Pontiac, Oldsmobile, and Packard the fuel pump has been placed below the operating lever to remove it from proximity with the exhaust manifold and to allow the bowl to protrude beneath the engine pan and be exposed to the outside air stream. The Terraplane-Hudson pump is provided with a glass trap after omitting it for two years. It has a capacity of 40 gal. per hr. at 1950 r.p.m. A vacuum booster unit is combined with the fuel pump as optional equipment. Most gasoline tanks have been revised to insure rapid positive filling without entrapping air. Packard provides corrugations across the top and bottom which stiffen the tank; also there is an air vent at the top and a sediment pocket at the bottom. The filler tube has been moved to the fender, taking the kinks out of the filler neck. The capacity of the Cadillac "60" and LaSalle tanks has been increased to 22 gal. On all Chrysler products the filler pipe has been moved entirely from the trunk compartment, permitting the lowering of the luggage-compartment floor. Studebaker has made its 18-gal. tank of the "flat" type for the same purpose.

A twin V-belt drive is used by Graham on its supercharged engines, the rotor traveling at 5.75 times crankshaft speed. Belt tension is maintained automatically. The Cord 125-hp. engine is boosted to 170 hp. when supercharged. Supercharging is being conducted experimentally on some of the General Motors bus models.

Electrical Equipment

Increased-output, current and voltage regulation of generators is becoming more common practice. The Delco-Remy shunt-type generator is used on the large Cadillacs, large Packards, and Studebaker President. The Delco-Remy split-field type is used on all Buicks, and the Cadillac "60" and "65". One field coil is connected to the third brush and the



Delco-Remy Automatic Choke Used on Buicks

other to the insulated main brush. The current and voltage regulator has been changed to include two resistance units, giving better voltage regulation in the higher-speed range. A 20-ohm resistance is connected across the current-regulator unit, and a 30-ohm resistance, across both the current- and voltage-regulator units.

Auto-Lite shunt-wound generators are controlled by a three-unit regulator comprising a circuit breaker, a voltage regulator, and a current regulator, enclosed in a single housing, permitting the control of line voltage directly at the generator. These units are each separately temperature-compensated, allowing them to function under all conditions at the values for which they are adjusted.

The Cadillac V-8 engine houses the generator in the engine vee. It is driven by the belt that also drives the water pump and, by using a 3½-in. pulley instead of a former 4-in. pulley, the low-speed range has been improved. A higher setting of the current regulator on the large models increases the charging rate above 20 m.p.h.

Battery capacities have been stepped up in many instances. A new development is the Kathanode method of construction in which the positive plates are encased in flexible and highly porous retaining mats composed of many layers of finely spun glass. Terraplane-Hudson locates the battery under the hood on the left side of the car. Cool air from back of the radiator grille alongside the core is conducted to the battery case. Willys likewise uses the under-hood location.

Cadillac has dropped vacuum advance of the distributor. It is now driven from the same shaft as is the oil pump, by the camshaft gear through an idler, giving all distributors clockwise rotation. The general design of the Oldsmobile-6 and -8 has been patterned after the 1936 six-cylinder engine with angular distributor and oil-pump drive. The Auto-Lite distributor governor-mechanism weights are pivoted from the top member and operate through a cam, allowing greater freedom of action and closer adherence to spark-advance requirements. A redesigned vacuum-control unit is of larger size and the diaphragm re-formed and of different material. The Delco-Remy vacuum-automatic type is made with two rigid ball-track ears and a third one of spring material, riveted to the breaker plate with a tapped stud to which the vacuum unit linkage is attached. A ball track is machined in the distributor housing, the balls being on the same level as the breaker cam, lever, and contact points.

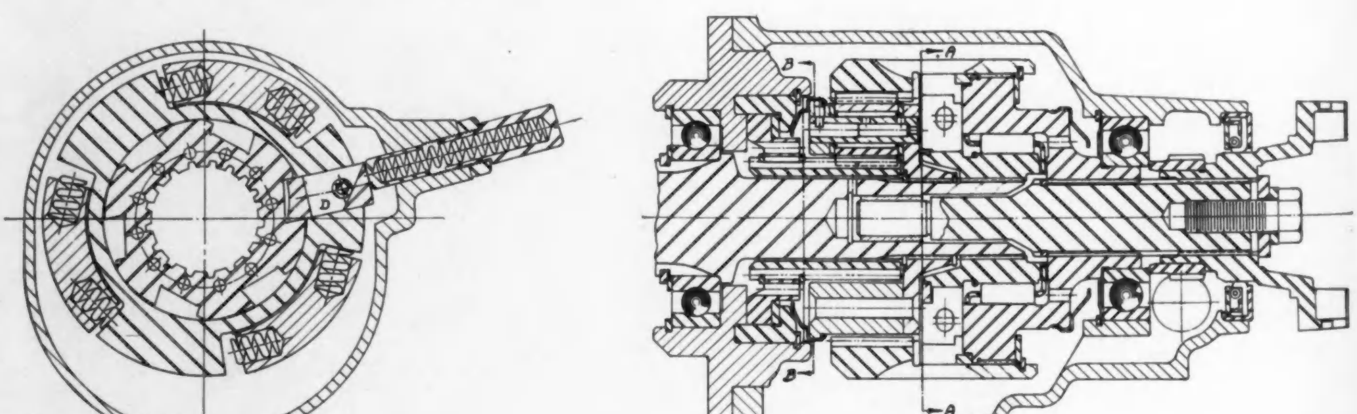
The 10-mm. spark-plug is used on all Packard engines. A special wrench is provided, and the plug is tightened by using only the fingers of one hand in order not to strip the cylinder-head threads. This AC plug is used also in the General Motors supercharged buses. Cadillac and Oldsmobile now have adopted the 14-mm. plug. Auto-Lite is producing a one-piece plug assembled by a mechanical "twist". The electrodes have a high electronic-emission factor, and the expansion coefficient of the cement conforms to variation of the wire and porcelain under any heat condition.

Engine Mounting

Cadillac reduces low-speed vibrations by the use of a three-point engine mounting, the two forward supports being carried on the frame side rails and the rear one carried below the transmission extension, the latter support being of the biscuit form with rebound cushion. The improved roadability of these cars is credited to this mounting with the wide spread of the front supports enabling the engine mass to be effective in steadying the front of the frame. The Buick "40" rear mounting has the rubber molded in one plane and at an angle; the larger engine has a smaller area to make it softer. Pontiac claims no noticeable torque reaction above 12 m.p.h. in high gear on the 6, and above 8 m.p.h. on the 8. The Firestone biscuit-type mountings provide various combinations of fabric for stability. New rebound units with restricted neck aperture hold down the fore-and-aft engine movement. The Packard-6 powerplant is mounted on a high front-center rubber cushion and two inclined rubber supports, one on each side of the transmission similar to previous practice. Nash and Lafayette use high front mountings and Firestone biscuits at the rear in their new three-point layout.

Clutch

Flexible motor mountings, livelier suspensions, and low-pressure tires have made the problem of clutch chatter a



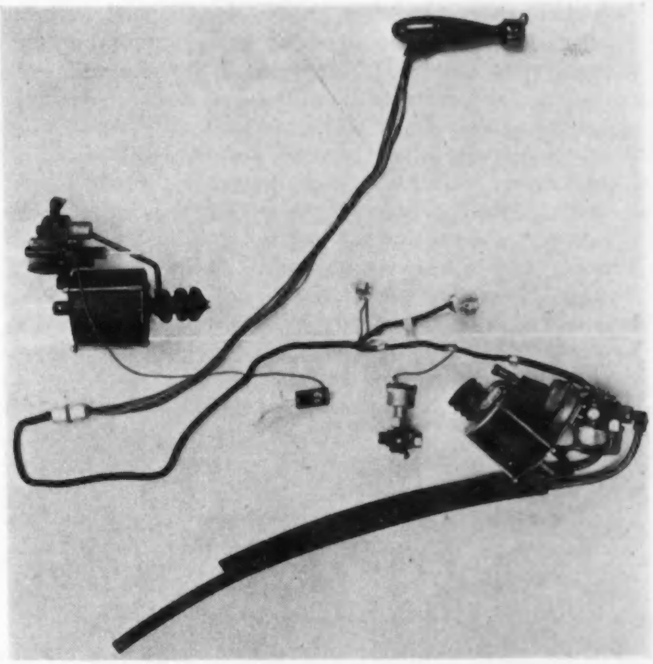
Studebaker Automatic Selective Overdrive

difficult one. The minimum cushion specification of the clutch-driven plate has been increased in a number of instances beyond previous practice. Whereas a 9¼-in. Borg and Beck compound-type cushion disc is used on the Lafayette with a minimum cushion of 0.042 in. with a maximum pressure-plate release movement of 0.060 in., the same diameter plate is used on the Oldsmobile-6 with a minimum cushion of 0.060 in. and a maximum drag of 0.080 in. The Oldsmobile-8 with a 10-in. compound-cushion plate has the same specifications. The standard 9¼-in. cushion plate is used on the Plymouth and Studebaker Dictator with 0.040 in. approximate cushion and 0.055 in. maximum drag. The Nashes, Dodge, DeSoto, and Chrysler cars with a 10-in. compound-cushion disc have 0.042 in. minimum cushion and 0.060 in. maximum drag.

Borg and Beck driven-disc assemblies are balanced by the use of clips of three different thicknesses to preserve a limit of 3 in-oz. and are laid on the disc in the proper angular position after the balance error amount and location have been determined in a balancing machine. Parallelism of the driven disc under load is measured in a fixture having six electrical contacts which light lamps when the error exceeds the 0.007 in. limit. Fully heat-treated thrust springs are generally used. The Oldsmobile clutch cover is provided with guide pins to take care of side thrust on the lever, caused by the graphite release-bearing ring. The pins also center the release-bearing plate more accurately. The Oldsmobile-8 flywheel rim has been removed, allowing a flat face similar to the six-cylinder design. Increasing the depth of the clutch cover has permitted larger ventilation holes in its periphery.

The crimped cushion plate in the Long clutches is made slightly stiffer at the outside to obtain a better wear pattern on the friction linings, with less tendency toward high-unit-pressure areas and also to increase the torque capacity by increasing the effective mean radius. The 9½-in. semi-centrifugal clutch is used in the Packard-6 and Graham has adopted this type throughout its line. LaSalle uses the 10½-in. and the Cadillacs, the 11-in. clutch except on the 16. New 7½-in. and 8½-in. models have been developed by Long but, since they normally require a low spring pressure, the semi-centrifugal feature is not required. The model "17", the largest Long clutch with a capacity of 600 ft-lb. torque, is used in the large General Motors Greyhound Coach. An endless woven facing of high bursting strength, of uniform rigidity throughout due to the absence of a joint and reduction in balance error, is used by Oldsmobile. The Buick "40" disc has been increased from 9½ in. to 10 in. and, the area from 82.8 to 100.5 sq. in. An air duct in the clutch cover plate and a vent hole at each side of the flywheel housing keep out oil vapors. The Terraplane-Hudson oil-cushioned single-plate type with heat-treated cork inserts lends itself to automatic clutch operation, which design is available as optional equipment. A rubber pad has been added on the clutch-follower mechanism actuated by the pedal to cushion its contact and cause quieter engagement. A round-section rat-trap type steel return spring is so positioned on the Oldsmobile that, during the first portion of its travel, it acts against the pedal tending to return it but, on passing over a center position, it acts in the direction of the pedal travel.

Permanent lubrication for the release bearing is provided on all Chrysler products by means of an Oilite ball retainer with the lubricant sealed in the bearing. A reservoir in the cast-iron retainer surrounding the graphite bearing of the



Terraplane-Hudson Automatic Selective Shift Mechanism

Oldsmobile release, is filled with non-liquid wax-base lubricant which is fed through the graphite by capillary attraction. The Cadillac transmissions as redesigned are lighter, and all gears are helical with the sliding low and reverse gear on helical splines. The shifter shafts are mounted on the side to permit lowering of the transmission top. The cover is on the bottom, and the shorter case increases rigidity. A three-pin type synchronizer similar to the Buick design is now used. The second-speed synchronizer-drum cam angle has been changed to 50 deg. on both the Buick and Oldsmobile. To facilitate the engagement of the sliding sleeve with the gear that is being synchronized, the chamfer on the sleeve of the Oldsmobile has been altered from being symmetrical to the axis of the sleeve to being symmetrical with the centerline of the helical tooth. The Buick synchronizer drums are bronze forgings. The detent springs of both Buick and Oldsmobile are thicker, wider, and their shape has been altered to prevent localizing of stresses. The Chrysler overdrive is now mounted in a separate unit bolted to the rear of the transmission case and centered by an adapter plate supporting the transmission shaft and into which the sun gear of the planetary train is pressed. A sliding lockout sleeve contains the slots into which the overdrive pawls engage, but into which they cannot register when it is shifted into the direct-drive position. A similar Warner R-6 unit is used by Graham. In the Studebaker automatic selective control in which the overdrive can be cut in or out at any speed above 35 m.p.h., centrifugal pawls control direct or overdrive, and means are provided for releasing the sun gear. The adapter plate contains a head into which the sun gear is pressed, and it is freed (at which time the drive is taken through the roller clutch) or locked stationary by a pawl sensitive to the coasting load reaction or to the inertia of an oscillating member surrounding the head and depending on whether the engine is again accelerated quickly or gradually.

Great interest centers in the Terraplane-Hudson automatic control. The Bragg-Kliesrath automatic vacuum-clutch con-

trol with inertia valve, accelerator plunger, and operating cylinder are used, but a solenoid valve permits automatic disengagement only when the solenoid is energized. This unit replaces the previous manual dash control. A high-gear lockout switch forms part of the selector-switch assembly and, by completing the solenoid circuit, permits automatic clutch disengagement while the selector lever is in "reverse", "low" or "second". A centrifugal governor is driven by the speedometer gears at the back of the transmission and completes the solenoid circuit when car speed is below approximately 17 m.p.h., thereby providing automatic clutch operation below this speed regardless of selector-lever or transmission position. A cutout switch on the instrument panel makes the automatic clutch inoperative when in the "off" position. The remaining hook-up is as in the previous models. While driving in high gear, the clutch will be fully engaged as long as the selector lever remains in "high" and the car speed does not drop below 17 m.p.h. If, however, the selector lever is moved to neutral or to any other position, the clutch will be disengaged and the selected gear change will take place the next time the throttle is fully closed.

Propeller Shafts

The Mechanics Universal Joint Co. has made an outstanding improvement in its joint in which a thin-walled, internally splined sleeve is pressed into, and welded in, the shaft tube and mates with an external spline on the yoke. The overhung weight is reduced, and the spline diameter is increased so that the lower unit pressures permit a lower hardness of the parts, enabling the machining to be done after heat-treating. Lubrication is improved, and better alignment of the tube and sleeve member obtained. As used by Cadillac, there are 36 splines on a 1.800-in. pitch diameter, 1.8795-in. outside diameter, 20-deg. pressure angle for the spline involute tooth form on the yoke with a 228-255 Brinell hardness number. The No. 1045 sleeve is similarly heat-treated. In the Oldsmobile application, a 32-spline, $1^{11}/_{16}$-in. diameter is used.

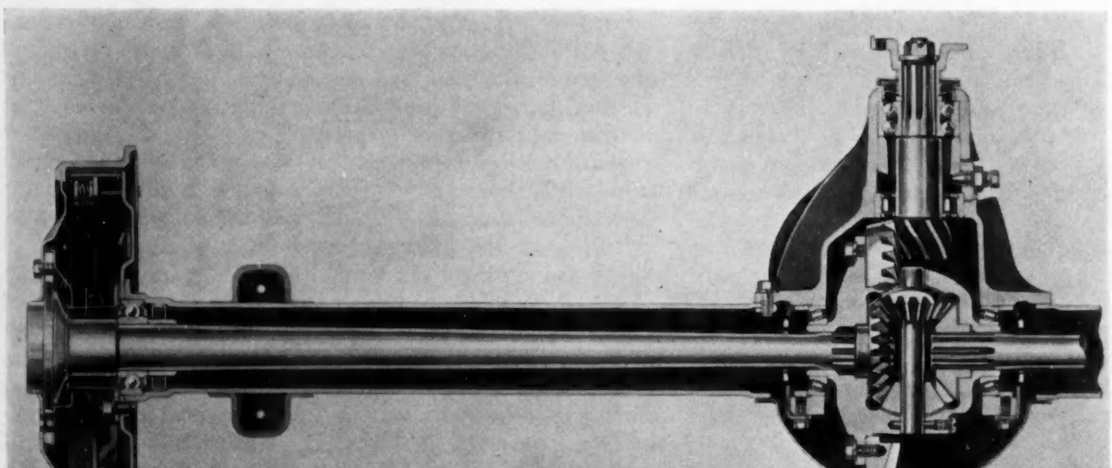
The Universal Products joint is made with a double boot, the inner part of which is of heavy molded leather, permitting a larger amount of lubrication and assuring its better distribution. It is used on Plymouth, Dodge, DeSoto, Chrysler Royal, and Graham. This company has developed a compact double joint for use as front-wheel drives, utilizing two ball-and-trunnion type joints with a center member, permitting accurate bisecting of the angle at all times. The Bendix-Weiss constant-velocity joint is used at the wheel end of the Cord, permitting a maximum angle of 37 deg. each side of center. A new Weiss joint has the wearing surfaces imbedded in a die-cast yoke as inserts. The Thompson Products joint, consisting of four rubber spools encased within two pressed-steel members, is used by Studebaker.

To lower the floor without resorting to hypoid gearing, a two-piece, three-joint propeller shaft is used by Oldsmobile. The center bearing is a permanently lubricated ball-bearing enclosed in a rubber collar and supported by the frame X-member in a position corresponding to below the rear of the front seat. The spline is incorporated in the rear joint, and the movement limited to the rear shaft permits an almost flat rear-compartment floor. Although the cost of a two-piece construction is slightly higher than a single shaft, the shorter lengths permit relatively smaller diameter tubular shafts. The short shafts raise the whipping period beyond any usable range. Pontiac, now with Hotchkiss drive, attacks the problem with a single two-joint shaft to the spiral-bevel axle. A fixed extension similar to Cadillac's is secured to the rear of the transmission in order to cut down the overall length of the shaft. An extension shaft has a slip-joint at the rear of the transmission consisting of an internal-external gear tooth coupling to allow for slight misalignment in addition to the required sliding action due to rear spring movement. The bearing supporting the rear of the extension shaft is a special bronze bushing protected at the rear by a slinger and felt seal. The coupling, bushing, and speedometer drive are lubricated from the transmission.

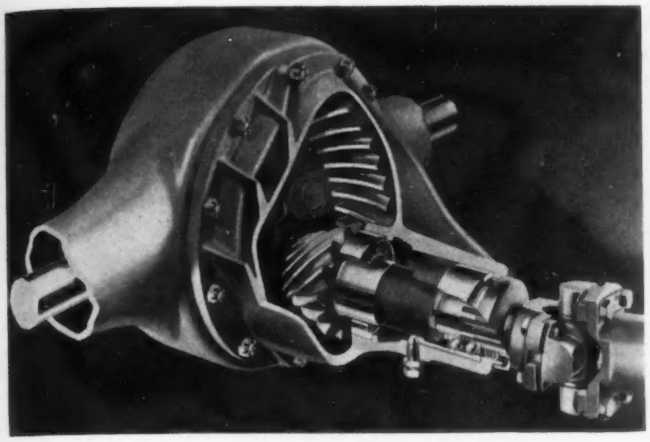
Rear Axle

Hypoid gearing has now been adopted by Chevrolet, Plymouth, Dodge, Chrysler (except on Airflows), Buick "40" and "60", Studebaker, Packard-6, Cadillac "60", and LaSalle.

The Packard "120" has changed from ball bearings on the pinion shaft to Timken bearings. The construction, also used on the 6, consists of a front and rear bearing for the overhung pinion. The rear bearing width is held to a tolerance of 0.001 in., and the dimension from the cross-bore to the shoulder in the pinion housing as well as from the cross-bore



Pontiac Rear Axle with "Non-Generated" Ring-Gear Tooth, Hyatt Self-Aligning Differential Case Bearings and Axle Shaft with Wheel Flange



Cadillac "60" Hypoid Axle

to the face of the pinion is held to 0.001 in., making a total overall clearance of 0.003 in. so that no adjustment is necessary for pinion location. The old type of solid spacer and shims between the two bearings is obviated by the use of a special spacer. The nut holding the companion flange is tightened in a special machine to a definite pre-load of 25 in-lb. on the bearing, after which time the spacer automatically maintains it. When running, the thrust load in either direction is taken directly through the bearing against its shoulder. The axle now has a welded-on cover.

Spicer is furnishing the Studebaker hypoid axles. The rear-wheel treads of the Chrysler Royal and the 121-in. wheelbase Chrysler Imperial have been widened to 60 in., while that of the Custom Imperial is now 63 in.

The Oldsmobile and Pontiac axle-shaft splines on the inner end have been changed from straight side to involute form; Hyatt self-aligning "barrel" roller bearings support the differential carrier; and the pinion shaft is mounted in a Hyatt roller bearing adjacent to the pinion and a standard New Departure double-row ball bearing at the front. The ring gear is of increased section and is bolted to the differential case instead of riveting. The wheel bearing is protected by two circular shields in addition to the integral bearing seals. A rawhide seal is located just inside the wheel bearing which is now closer to the centerline of the rear wheel through the use of an upset wheel-supporting flange on the outer end of the axle shaft, replacing the previous tapered hub. The Oldsmobile spring seats are 45⅜ in. apart instead of 43 in.

The Pontiac ring-gear tooth has no curvature in the normal plane but is a rack tooth. Conjugate tooth action is secured by increasing the curvature (in the normal plane) of the pinion tooth. Greater accuracy of tooth form and better surface finish are the advantages of the "non-generated" type of ring gear. Nickel-molybdenum steel is used in the gear and pinion. The Buick, Cadillac "60", and LaSalle use a lower spiral angle than is ordinarily employed on hypoid gear teeth, to reduce bearing loads, to insure better tooth contacts, as well as to secure less distortion during heat-treatment. In the redesigned Lincoln-Zephyr axle, the radius rods extend farther out to increase the load-carrying capacity with less distortion. Buick also is using the upset wheel flange on the Buick "90" axle and the propeller shaft is supported at the front end by a bronze bushing. The Buick torque-tube ball head is tin-plated, whereas the stationary members are copper-plated. A plain cork seal is replaced by a Duprene bonded-cork washer pressed against the head by a Belleville spring washer. The Eaton axle has a positive lubricating system consisting of a drum or flange affixed to the ring gear and which submerges below the oil level. A scoop at the upper side of the housing scrapes oil from it which then flows by gravity through passageways to the pinion and differential bearings. To avoid hypoid gearing Briggs Mfg. Co. obtains an equivalent permissible floor drop by inclining the spiral-bevel pinion shaft downwardly toward the front and attaching the torque tube at an angle to the sloping pinion-carrier housing. A Bendix-Weiss universal joint can operate up to 18 deg. maximum fixed angle and, of course, maintains uniform velocity. Columbia Axle has a two-speed vacuum-shift unit for the Lincoln-Zephyr.

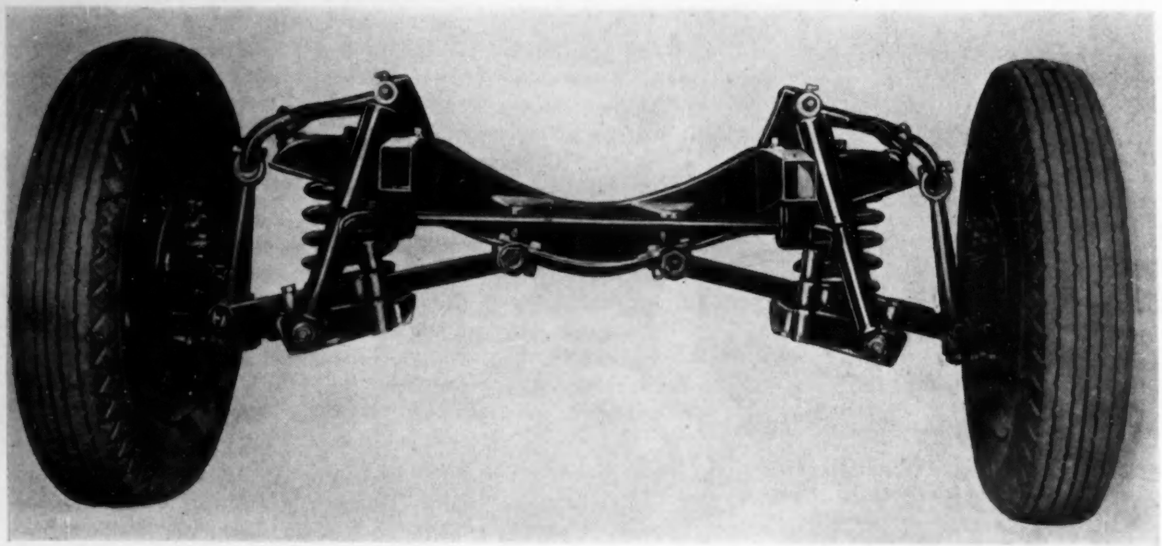
Brakes

The automatic "hill-holder" is optional on Terraplane and Hudson. A ball-ended rod applying pressure to the Buick master-cylinder piston reduces the chances of binding. A cover is provided in the floor over the cylinder for inspection and filling. Backing plates and shoes are zinc-plated. The Cadillac-16 and Packard-12 have a vacuum booster combined with the brake-pedal assembly unit. Oldsmobile is using Centrifuse drums on the 8 and the Kelsey cast-in-back type on the 6. The lining width has been reduced from 2 in. to 1¾ in., and eccentric anchor-pin adjustment is provided. The hydraulic front-wheel cylinder pistons have been increased to 1³⁄₃₂ in. diameter, whereas the rear-wheel and master cylinders remain 1 in. to secure 55-45 per cent braking. A new brake seal is made by welding a flat flange to the backing plate fitting closely to the drum which, furthermore, is surrounded on the outside by the offset out-turned periphery of the backing plate. Buick has eliminated the hand-brake cross-shaft and substituted a triangular plate with three holes, the forward one taking the hand-lever cable and the other two taking the rear-wheel cables which extend diagonally back. Although some of the hand levers are still somewhat crude, a number of refinements have been made in the form of comfortable pistol grips. The handle of the Oldsmobile lever has a molded grip conforming to the contour of the hand. The release lever partially surrounds the handle and, in extending directly to the pawl, eliminates the ratchet release rod. Rockwell Products "Electroflo" system comprises an accumulator with an electrically driven pump which maintains fluid therein under a constant pressure of 450-500 lb. per sq. in. The pump automatically cuts in when the pressure drops. A spring-loaded piston maintains the pressure in the accumulator, and a valve connected to the foot pedal feeds the fluid to the operating cylinder.

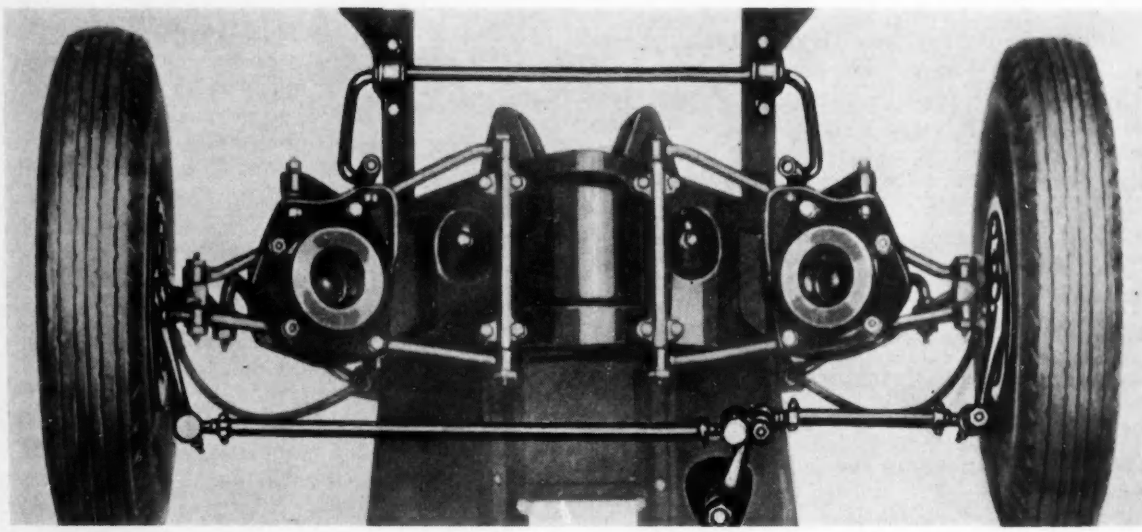
Bragg-Kliesrath has developed a universal relay valve, permitting tractors and trailers having dissimilar vacuum power-brake installations, atmospheric or vacuum-suspended type, to be operated together and have their brake operation synchronized. A simple camp-trailer vacuum brake utilizes a transversely mounted floating cylinder connected to one brake cable and its piston to the other.

Front End

The Terraplane and Hudson radius arms have been shortened slightly, and the axle connection has been modified to eliminate the pivot pin by substituting a solid connection. The Lincoln-Zephyr I-beam has been made heavier for strength and the caster changed for easier steering. Oldsmobile has reduced it from 1¾ deg. to ½ deg. On the heavier Cadillacs the caster angle is approximately zero. Such



Chrysler Front Suspension and Direct-Acting Shock-Absorbers



Chrysler Parallel Anchorage of Suspension Control Arms and Direct Drag-Link Steering Connection

changes increase the responsiveness of the steering mechanism to any movement of the steering wheel, return it more easily to center position, and decrease the tendency of the car to react unfavorably to cross winds. The DeSoto king-pin angle has been changed from 9 deg. to 4½ deg. Larger front-wheel bearings are used on the Cadillac "60" and LaSalle. Due to the reduced weight of the Cadillac "65" and "70" the outer and inner front-wheel bearings are smaller.

Wheels and Tires

Conical-disc wheels with 16-in. rims and larger hub caps prevail. Motor Wheel is supplying the conical disc to Chrysler, Packard, Buick, Hudson, Cadillac, Graham, and Auburn. Its Oldsmobile wheel features 6 wide flat spokes. The Kelsey Chevrolet wheel is of similar appearance, but with 8 spokes. The Dodge wheels permit the use of "slip-on" chains. Pontiac provides a shallow hub cap. In the majority of cases caps run from 9½- to 10-in. diameter and drop-center rims are used 100 per cent. Right- and left-hand threaded bolts have been adopted by Oldsmobile to obtain a self-tightening effect.

Tire sections remain about the same, the only new size being the 5.50-16 for light models such as Willys and the small-displacement Ford. Chevrolet has changed from 5.50-17 to 6.00-16. The Packard-12 uses 8.25-16. The General Motors Greyhound buses take 10.50-18 tires, this being the first time that a large-section tire has been put out in volume on an 18-in. rim. Tire pressure is carried at 50-55 lb. per sq. in. as compared with 65-70 lb. per sq. in. in the past.

Suspension

More powerplants have been moved farther forward for better weight distribution. Chevrolet has done so and entirely redesigned it for greater compactness permitting relocation of the body, radiator, and the hood. The Cadillac "60" and LaSalle rear-spring leaves are separated by waxed liners. The leaves have grooves on the under side, and no covers are used with this type. Both these cars have threaded shackle pins and spring bolts. The other Cadillacs have similar shackle bolts, but the front-spring eye bolts are rubber-bushed. The Nash spring leaves are pre-lubricated and fitted with impregnated bronze inserts.

Packard has adopted the Berlin-type spring eye on all models. The Pontiac silico-manganese rear springs have a deflection rate of 126 lb. per in. for the sedans. The Lincoln-Zephyr is equipped with a transverse radius rod on the left

side, paralleling the steering drag link, holding the axle in definite relationship to the body. It permits the use of longer, softer springs with normally vertical shackles. A similar transverse stabilizer is used at the right rear between the axle and body, both materially improving the steadiness of the car in strong cross winds and on rough roads. Heavier shackles and larger pins are used. The 14-leaf front spring has a total rate of 300 lb. per in., and the 15-plate rear spring, 250 lb. per in.

Front and rear stabilizers, introduced last year by Cadillac, have been extended to Buick and Oldsmobile, the former with the unit behind the front suspension and the latter with it ahead. The Cadillac "60" and LaSalle have a rear stabilizer consisting of a link extending crosswise behind the rear axle. The right end is secured with a ball joint to a bracket on the frame cross-member, whereas the opposite end is connected with a second ball joint to the left-hand spring pad. This construction counteracts both vertical and sidewise chassis and body movements. The Monroe stabilizer is used on the Terraplane, Hudson, Chrysler, DeSoto, and Nash.

Individual Wheel Suspension

The Packard Safe-T-Flex system has been applied to the 12 and Super-8. Pontiac is now using the parallel-link type, permitting the deflection weight at the spring to drop to 90 lb. per in. as against a corresponding 190 lb. per in. on the 1936 Master-6, while the oscillation rate had dropped from 100 to 72 per min. The Chevrolet springs have been entirely redesigned due to the decreased weight and relocation of the body and chassis units. The lower control arms on the Oldsmobile and Buick "40" and "60" have been changed from a square section to an I-beam section.

The DeSoto, Chrysler Royal, and Imperial control arms are secured to the frame by parallel anchor rods. With the adoption of direct-acting shock absorbers, the upper control arm is supported in a plain bracket.

Frame

The generally lower floor levels have brought about more double-drop frames. Chevrolet is using a more rigid frame of the box-girder type introduced on last year's Standard. The A. O. Smith I-beam X-member has been a distinct contribution and is used by Pontiac, Oldsmobile, Buick, Cadillac, LaSalle, and Packard. Increased rigidity ranges from 120 per cent and 80 per cent on the Oldsmobile-6 and -8 respectively to 430 per cent on the Packard. This improvement has resulted in a better foundation for the body as well as in increasing the road stability of the car.

Equipment

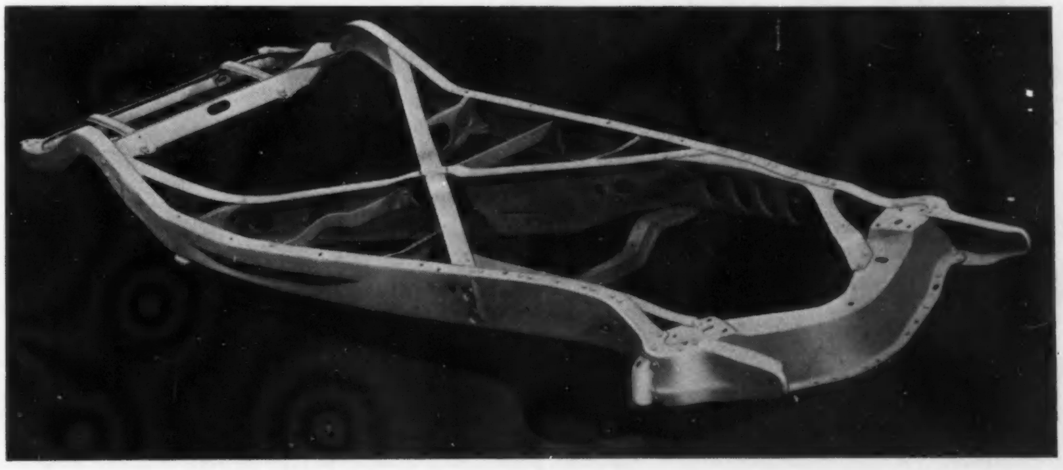
Wider single-bar bumpers are being used, and bumper guards are generally standard, often incorporating a monogram or emblem. Oldsmobile provides an ornament at the center of the front and rear bumpers to distinguish the two models. The mounting bars pass through the sheet metal at the front and rear with rubber grommets sealing the opening. A bar is welded to the rear spring shackle and extends rearward to provide a jack pad. Studebaker supplies bumper jacks.

Outside horns are disappearing. The Lincoln-Zephyr horns are mounted under the hood top just forward of the dash for better tone and engine accessibility. On the Cadillac V-8 and LaSalle horns, the shape of the projector has been changed and they are mounted on the air cleaners. Auto-Lite now has a die-cast conch-shell type projector. Whenever horns are supplied for matched installation, the pitch is determined accurately by means of an electric stroboscopic-calibration device whereby they may be selected to match within 1½ cycles of pitch.

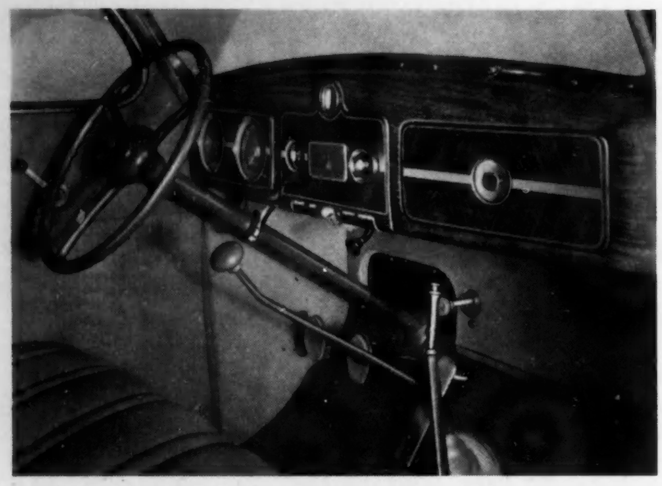
Long bullet-shaped head lamps prevail, and they are set lower in the case of Pontiac and Oldsmobile. The Buick lamp shell is faired into, and mounted on, the hood side. The fender lamp is mounted on the same level as the head lamp and is lit for parking or when the headlight beams are partially or fully depressed.

At the rear-center of the General Motors cars a light is mounted above the license-plate support and on the rear-compartment lid. On the Chrysler products the lamp is below. Nash has placed a light below the trunk opening so that it will also illuminate the trunk interior. The Oldsmobile combination tail and stop lights are located high on the body at the termination of the chromium body molding. The 6 has a streamline shell similar to a miniature headlight reversed; the 8 has a square modernistic design with both rear and side panels of red glass. The Hudson and Cadillac tail-lights are on the rear fenders. The Cadillac tail-light body and support is a single die casting, covering by a second unit.

A combination tail and stop light bulb has been introduced by the General Electric Co. in which the filament arrangement produces a wider differential in the tail and stop indications. To prevent improper insertion, a new indexing base



Oldsmobile I-Beam X-Type Frame



Plymouth Instrument Board with Recessed Controls and Knobs

is used, similar to the two-pin bayonet except that one pin is ⅛ in. higher than the other, and the socket has one short and one long J-slot. It is used by Cadillac, LaSalle, Buick, and Oldsmobile.

With the increased use of voltage-control shunt generators, there is a trend toward an indicator lamp to replace the ammeter, such as is found in Pontiac. The trip speedometer on the Graham custom models is provided with a re-set knob on the face of the instrument panel. The previous ruby tell-tale lights used by Terraplane and Hudson to indicate oil pressure and charging are now small semi-opaque panels. The word "no" flashes in red when there is no oil pressure and the word "not" when the generator is not charging. The dials of the Terraplane and Hudson instruments are black with the figures in silver finish. The Graham edge-lighted instruments with numerals etched directly on the glass are grouped in a vertical center "tower section" in the panel. Instrument grouping into one unit is generally favored and occupies the left side of the instrument panel with the speedometer. Color schemes for panels and instruments are many and harmonize with the body interior. Horizontal chromium strips or beads across the panel are favored.

All Chrysler products except the Airflows have all controls including the ignition key recessed as a safety measure. The windshield wiper controls are soft rubber instead of metal. The opener control folds from view, replacing the protruding crank. The Lincoln-Zephyr panel includes a centrally enclosed grille reaching down to the top of the transmission housing to screen the installation of a heater and radio. The Buick and Oldsmobile panels provide a die-cast center-grille section for the radio. The Pontiac and Studebaker Dictator speedometer dials have the numerals on one horizontal level with radial lines for the speedometer hand to follow. Package compartments have been increased in size and, when an electric clock is installed, a small light illuminates it on the Oldsmobile and Buick when the door is opened. The push-pull type light switch is standard on many instrument boards, sometimes with a high headlight-beam indicator in its center in place of the popular speedometer tell-tale.

Most cars provide for a hot-air windshield defroster, designed to attach to the top of the hot-water heater. A small centrifugal blower forces a portion of the warm air through two tubes where the air is directed against the bottom of the glass at each side. Buick and Oldsmobile make use of the car running board as an aerial by insulating the brackets from the chassis frame. Dodge makes provision for the installation of an extra radio speaker in the back of the front seat of four-door sedans.

Direct-acting shock absorbers are used by Studebaker, Chrysler, DeSoto, Dodge, Plymouth, Terraplane-Hudson, and Willys. Several revisions have been made in the Oldsmobile shock absorbers to insure quiet operation, consisting of a non-rotating piston and the adoption of angle-type valve seats. Fixed-type, 1½-in. end-to-end discharge shock absorbers are used on the Cadillac "60" and LaSalle, front and rear. The larger Cadillacs use 1¾-in. units, and the rears have clicker adjustments and dash-pot inertia controls. The Lincoln-Zephyr rear shock absorbers are located under the frame rails, where they are accessible for inspection or filling. The Buick and Oldsmobile shock-absorber links are provided with a rubber insulator molded to the link stud and retained in the link by rolling the sides of the link end around the insulator at assembly. The stud has a taper fit in the shock-absorber arm.

Lincoln-Zephyr Instrument Panel with Lower Extension Housing Heater and Radio

Control

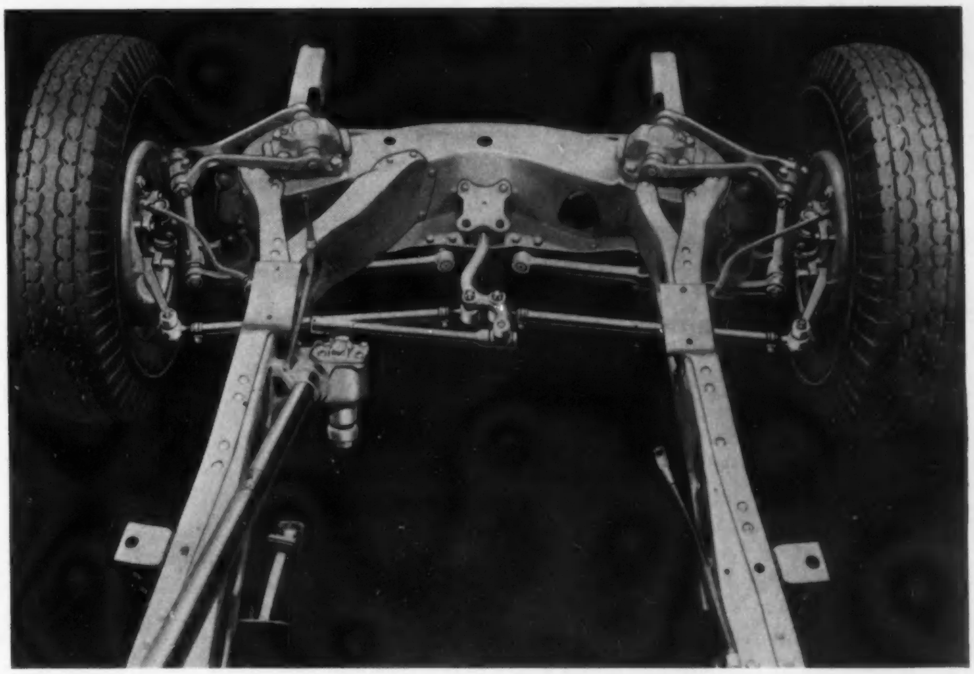
The Ross twin-lever steering gear is used by Studebaker and the larger Graham models. Gemner and Saginaw are using straddle mounting of the steering-arm shaft for greater rigidity. In the Saginaw gears the worm end-play is adjusted by a set-screw at the bottom, and roller lash is adjusted by rotating the worm into the roller by means of an eccentric housing. Instead of running the horn wire through to the base of the column, it is taken to a point below the dash where it emerges through a hole in the steering tube and is soldered to an insulated metal sleeve. Current is conveyed to a plate spring acting as a brush and secured to the outside column tube.

All of the General Motors cars except Chevrolet use a transverse drag link, eliminating the previous ball-crank center lever and transferring the steering force directly to the drag links. These drag links are now provided with a spring on each side of the ball studs. In the DeSoto and Chrysler individually sprung fronts, the steering-gear assembly is farther forward on the frame and is coupled to a short left drag link and a long right link. With the parallel pivoting of the control arms, the theoretically ideal center-point relationship has not been found to be essential.

The Tenite steering wheel and spring-rod type spokes are very popular. In the Lincoln-Zephyr the color of the wheel and the column has been changed to chocolate brown for more warmth. The ratio has been changed to 20.2:1 from 18.1:1. Terraplane and Hudson have changed from 17:1 to 18.2:1 and increased the steering-wheel diameter from 17 in. to 18 in. Pontiac has stepped up its ratio from 17.5:1 to 19:1. With wider seats, Pontiac has moved the steering wheel to the left, allowing more room for the middle front-seat passenger. Dodge has decreased the ratio from 18.2:1 to 16.4:1.

Grilles and Sheet Metal

The horizontal radiator grille of convex form predominates and is especially effective when it is carried back in integral formation with the hood louvres as in the case of Studebaker and DeSoto. Buick features a two-piece horizontal grille, vertically adjustable at assembly, separated by a tapering body-colored center section which is a continuation of the tapering hood top panel. Pontiac provides a center six-bar grille extending from the bumper to the cowl with a horizontal grille of five groups, the upper one continuing back on the hood to form louvres. Terraplane and Hudson maintain their grille design, but it is now flanked by horizontal louvres at each side. An increased taper characterizes the front slope of the large Packards. Chrysler provides a convex grille with a relatively low top edge and thin-section die-cast hood louvres, approximately 56 in. long and weighing 3 lb. each, that almost meet at the center of the hood top. A distinct note in the Chrysler fronts is the rounded formation which contrasts with the prevailing sloping straight front centerline.



Typical General Motors Steering Layout with S-Center Lever. Cadillac "60" Is Shown

The Oldsmobile-6 provides a bold horizontal design with eight die-cast grilles with three V-louvres in the apron below. The Oldsmobile-8 die-cast grille is of square-mesh design with corresponding hood side louvres. The General Motors cars have a vertical vane on the inside of the grille to straighten the air-flow.

The one-piece rear-pivoted hood top introduced by Lincoln-Zephyr is used in the Studebaker and Chrysler with the ornament acting as a hood latch. Chrysler provides an additional safety catch on the extreme front end which must first be released before raising the hood. In place of a grille, Willys provides a one-piece hood with a decidedly rounded front replacing the radiator grilles with louvres stamped therein permitting air to circulate through to the core. The third, fourth, and fifth louvres extend back along the sides of the hood. The longitudinal louvre-bar design predominates and conforms in some way with the general grille scheme. The Nash stainless-steel hood strips are for decoration only and do not serve as louvres. The Chevrolet hood extends over the radiator core to the grille, eliminating the shell. Sides are recessed back to clear the headlights and are riveted to form a rigid unit with the top half. The Pontiac hood side is held in place with wing nuts and is easily removable. Turning the hood catch mounted on the hood side raises the top half

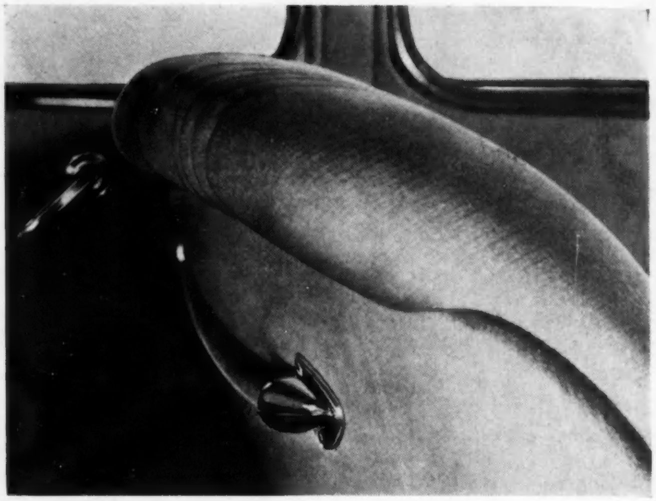
so that it can be grasped without the need of a hood handle. The Oldsmobile side panel is held in place by four bolts and extends from the grille to the cowl with the headlights mounted on it and removable with it. The hood catch functions as in the case of Pontiac. The Studebaker hood side has an inside fastener to permit its removal.

Fenders are generally deeper and wider and, in the case of Terraplane and Hudson, the wheel opening is considerably smaller than is customary. In the Packard the front apron now has a smooth flowing line that blends into the fender. The lower inside portion of the Pontiac aprons has a raised panel effect which quickly fades out. A similar raised design is used on Chevrolet but follows along the apron to form a shelf which continues back through the cowl side and the major portion of the front door, giving a "wind-streak" effect. A single "wind-streak" rib is found in the Pontiac fenders. The Willys fenders provide an "eyeball" headlamp which is faired into the fender crown in an elongated-barrel form. Nash provides a pointed chromium decoration at the center of the creased fender, extending fore and aft of the parking light at the front and the tail light at the rear. A matching strip is located on each front apron. The rear-lower corner of the front Oldsmobile fender is flared horizontally, whereas the Buick and Pontiac back-end has been raised in Cadillac fashion.

Body

The all-steel body is used in the entire General Motors line except the Buick "80" and "90". The Lincoln-Zephyr now includes a three-passenger coupe. Bodies are considerably wider, giving more seating room and a wider windshield. In a number of instances the bodies are also longer. Side windows due to the lowered floors are located higher and provide better vision. Oldsmobile has moved the body 6 in. forward in relation to the rear axle. It has a new club coupe in which there are two transverse folding seats behind the driver's seat. Windshield openings are higher to improve visibility. Oldsmobile and Pontiac have increased the slant from 31 to 39 deg. and Chevrolet from 31 to 35 deg. The Packard-6 and

Chrysler-Built Bodies Are Mounted on Rubber Spools Supported by Short Outriggers



The Dodge Padded Front-Seat Edge Incorporating Sponge Rubber

"120" have foot-high windshield openings, and 40 per cent more powerful double windshield cleaners are used on the large models. V-windshields are used on the Chrysler Airflows only. There is no DeSoto Airflow. The Packard "120" convertibles use die-cast windshield frames with overall dimensions of 50 x 22 in. and that weigh around 29 lb. Cord uses a similar-type casting weighing about 30 lb.

In changing to an all-metal sill, the Fisher bodies have lower sills and floors. At the rear door, the Oldsmobile sill has been lowered 3½ in., reducing the step to the running board. The Pontiac rear-compartment floor has been lowered 3 in. so that it is now 15½ in. from the ground. Chevrolet's rear-floor height is 16⅞ in. from the ground, which is 2 in. less than last year. Hudson and Terraplane have a 2-in. lower height accompanied by 1½-in. lower center of gravity. The Buick "40" and "60" floors have been lowered 1¾ in. In general, all front-compartment tunnels are lower.

Wider rear-door openings are prevalent and heights, front and rear, also have been increased with the lowering of the floors. The wider rear doors have made it possible in many instances to lower the rear-door windows out of sight. Terraplane and Hudson rear doors are 4½ in. wider and the front and rear doors, 1½ in. higher. The Pontiac front-door openings are 1½ in. higher and the rear door 3 in. higher and 5 in. wider at the floor level. Packard and Fisher bodies among others have a sponge-rubber seal around the edge in place of the previous extruded-rubber strip. The resistance of the sponge-rubber seal permits the use of less tension on the door dove-tails. Rear windows are universally of the full-length, vertically pivoted type.

The one-piece roof is now used on all Chrysler products except the Airflows, which retain the inserted-center steel-panel section. Graham also uses this construction. Happily, drip moldings have returned, being welded to the in-turned flange of the top panel. Center posts are invariably of box section with a wide flare at the top. The divided window center strip in the rear panel is now an integral portion of the rear-panel stamping, whereas it was welded in on many of last year's body shells.

The Fisher bodies have a lower side panel curving inwardly to the belt molding with the sides sloping inward above it. The lower panel is carried forward through the

hood to the radiator grille. The Chrysler roof, upper-rear quarter, and cowl sides are covered with Wafflex insulation consisting of two layers of treated paper with a wadding material between. Silento is used on the side panels. The steel floor is covered with insulating material which adheres after the body has passed through the baking oven. The Buick "40" and "60" roof, side panels, and trunk compartment are coated with special black insulating paper, whereas the "80" and "90" have a hair-felt blanket. The General Motors Greyhound buses have achieved remarkable absorption efficiencies by applying a special blanket composed of light-weight mineral wool affixed to the metal roof. A perforated membrane of Johns-Manville Hardboard is supported in grooves in the base of the die-cast purlins, with a space between it and the blanket through which ventilating air circulates. The Hardboard is perforated with $\frac{1}{8}$-in. holes on approximately $\frac{9}{16}$-in. centers. Sound waves that get into the bus pass through the perforated board and are absorbed in the wool. Such measures are, of course, not yet feasible in passenger cars where the problem of cost is vital.

The bodies of the Chrysler products, other than the Airflows, are mounted on short frame outriggers which incorporate a pure gum-rubber spool. The body bolt passes through the center, and a steel spacer prevents the bolts from being tightened beyond a predetermined compression. There are fourteen such mounting points, and the floor is furthermore fastened to the frame X-member at four points.

Studebaker running boards are colored to match the body. Pontiac and Nash boards do not join with the rear fenders. The Nash board continues forward to the back of the front-fender wheel-opening.

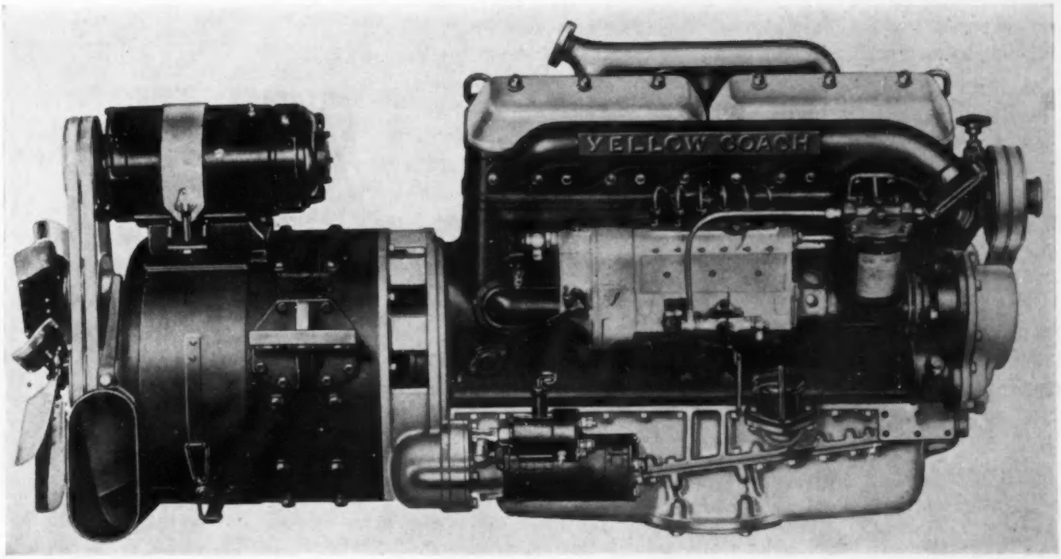
Buick uses Tenite handles for all door, window, and instrument-panel control fittings. The Studebaker steering wheel, horn button, and gear-shift knob match the instrument panel and package-compartment door. The Cord steering wheel is the same color as the exterior of the car as are the door knobs, window-regulator handles, and instrument-panel background. Fisher bodies have dropped the toggle interior lock and reverted to the button type. Door handles on all makes are of the safety type with both ends curved inward. The Mitchell Specialty Co.'s door dovetails are now used on the Packard-6 and "120". All Packards are equipped with the concealed upper hinge introduced on last year's "120". Rotary door latches have been introduced by Studebaker. Dodge, Chrysler, and Willys place a reading lamp above the rear windows which supplants the usual dome light. Packard, DeSoto, and Chevrolet (the latter in the Deluxe accessory group) use the Casco Products cigar lighter which, after heating to the right degree, automatically cuts off.

Most of the adjustable front seats have a 4-in. forward travel and, during this movement, the back of the seat is raised $\frac{3}{4}$ in., tilting the back of the seat forward. A spring assists in moving the seat forward. A carpeted recess at the base of the front seat to form a foot rest for the rear passenger is used generally. On the Hudson and Terraplane four-passenger coupes and broughams, the seat is pivoted at the left end and can be rotated forward in its entirety around the pivot point.

As a safety measure, a generously padded rolled edge is used on the Plymouth front-seat top and, in the other Chrysler products, the padding is supplemented with sponge rubber to protect the rear-seat passengers should they be thrust forward. As a further safety precaution, a silk robe cord is used instead of a metal rail. The Fisher bodies are provided with a shelf back of the rear seat. The Buick "40" and "60" sport coupes use two opera or auxiliary seats which fold completely out of the way when not in use into recesses in the bulkhead between the passenger compartment and the rear deck. They are hinged at the bottom and provided with a single-strut support which folds against the seat bottom.

Pontiac offers a choice of light-tan lustrous mohair or mixed-wool upholstering cloth. The convertible coupes are trimmed either in buffed, crush-grain, tan Spanish leather, or taupe-colored worsted Bedford cloth. Studebaker, on the tan and dark-colored cars, uses blue-grey upholstering, and tan fabric in the light colored cars. The Packard-6 is finished in broadcloth or barkweave. The large Packards have a tufted-pillow type upholstery in broadcloth. Nash provides a broadcloth with a four-paneled design. The Cadillac "60" offers a choice of tan or grey Bedford cords, or ribbed or plain broadcloths. When cords are selected they are used throughout in the trim, cushions, arm rests, and seat backs. The Cadillac Fleetwoods have a tufted-trim mode.

The increased sloping rear panels have augmented the luggage space which, in the case of the Buick "40" and "60", has been increased from 9 to 12 cu. ft. on plain back models and



General Motors Bus Powerplant with Hercules Diesel Engine and Generator of GE Electric Transmission

Flowing, Sweeping Lines Prevail, as Exemplified by Dodge

from 9 to 14 cu. ft. on trunk-back models. The luggage compartment of the Studebaker custom sedans, including space for the spare tire, has 17.4 cu. ft. capacity, whereas the cruising sedans have 22 cu. ft. A light inside the lid illuminates the compartment at night. Most luggage compartments provide for the placing of the spare wheel and tire at the bottom with a shelf extending entirely across the compartment in most instances, and only over the tire in some models allowing the remaining full height to be utilized. The placing of the spare tire in coupes behind the passenger seat is now used by Studebaker and the Chrysler products. In the Lincoln-Zephyr, the deck lid is lifted whereupon the spare tire can be swung down out of the way, giving access to the luggage space, which is no longer accessible from the interior of car. Packard and Fisher bodies are provided with a trunk-lid latch which automatically releases when slightly raised prior to closing.

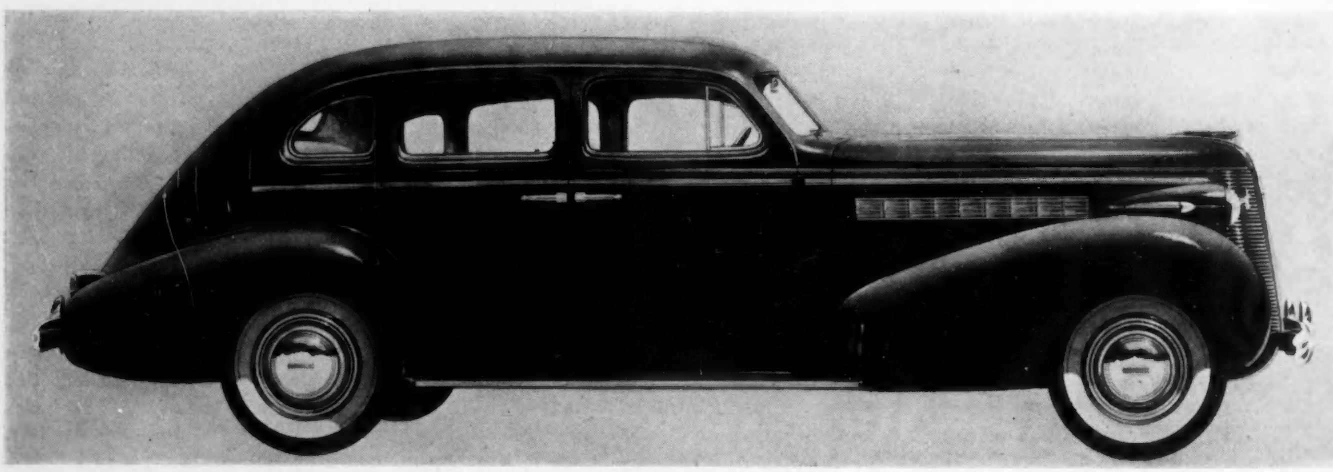
Heavy Vehicles

The General Motors rear-engined Greyhound interurban bus with the seat floor above the wheel houses and a depressed center aisle is one of the interesting developments of the year. This company also has developed a double-deck city bus with a stairway behind the driver's seat, and seating 31 passengers on the lower deck and 41 on the upper. The Model "733", 23-passenger city bus weighs 6500 lb. ready for the road.

American Car and Foundry has developed an "H-16", 42-passenger bus with the front-entrance door ahead of the front axle and center exit, powered with the Hall-Scott 180-hp. horizontal underfloor engine. A heating and ventilating unit has been developed, taking in fresh air through louvres in the side at window-sill height, above the possible layer of carbon-monoxide gas, to a heater compartment located under one of the transverse seats. A 12-volt motor drives a DeBothezat pressure fan that forces the air through a radiator core capable of providing 1200 B.t.u. per min. and discharging 420 cu. ft. of fresh air into the vehicle in place of the usual recirculated air. Ventilation is provided in warm weather by cutting off the hot-water supply to the core.

The Eastern Massachusetts Street Railway has been operating a Diesel-powered "37 R" Twin Coach bus during the past year. The "40 R" Model is on test using the Hercules "DRXB" engine with General Electric transmission, which powerplant also is used in slightly modified form in the General Motors bus for Public Service. The characteristics of an electrical drive enable the Diesel to operate to best advantage. Hercules has produced two small Diesels of 260 and 298 cu. in. displacement, operating at 2600 r.p.m., with a 3-in. diameter crankshaft, a stroke of 4½ in., and a bore of 3½ and 3¾ in. respectively. The former is used by Marmon-Herrington in a Ford conversion unit.

The 2650-lb. Stutz-Pak-Age car is equipped with a 113 cu. in. displacement engine, has a maximum carrying capacity of 3600 lb., a maximum governed top speed of 30 m.p.h., and the car negotiates a 5½ per cent grade in high gear with a load of 2000 lb. The Lintern Corp. is producing both air-and electric-operated sanders for traction, the latter drawing a maximum of 9 amp. at 6 volts or 4½ amp. at 12 volts. The Wisconsin double-reduction axle provides two trains of constant-mesh helical gears after the bevel reduction. Either of the floating pinions is picked up by an internal-external gear clutch. To prevent the scoring of the hardened case on the bevel-gear teeth until it has been hammer-hardened in service, Timken Detroit has developed a Lubri-Coating process to cut down the breaking-in period on truck and bus axles. The Detroit Compensating Axle Co. has produced a "compensating" trailer axle in which the dual wheels are journaled independently on a spindle end, and in which the axle center is attached with a horizontal pivot to compensate for road crown.



The Buick "40"

Manufacture of Elliptical-Skirted Pistons

By E. S. Chapman
General Works Manager, Plymouth Motor Corp.

THIS paper describes production methods of pistons characterized by an elliptical skirt having a slot or slots that affect the form of the piston under operating conditions, as practiced at the Plymouth Motor Corp. Design changes that have affected production technique and changes in production practice itself are important recent developments in aluminum-alloy pistons.

Foundry data include a description of the small open-hearth furnaces and the permanent molds with collapsing cores employed. Other foundry operations – heat-treating, sprue cropping, hardness testing, and rough machining – are dealt with fully. In this foundry an output of 18,000 pistons per 24-hr. day can be maintained in an area of 1,200 sq. ft.

This type of piston fits the cylinder bore with about a 0.001-in. clearance in a cold motor, the expansion at operating temperatures being compensated for in the slotted skirt. This skirt is in the form of an ellipse whose minor axis is 0.011 in. less than its major axis. The skirt also is tapered, as indicated in the paper along with other design details.

Included in the description of the fourteen machining operations are equipment specifications, feeds, speeds, tooling, and production rates. Operation No. 13 is anodizing in which the piston increases its growth uniformly about 0.0003 in. as it is carried by conveyor through an electrolytic bath of sulphuric-acid solution.

DEVELOPMENTS in aluminum-alloy pistons during recent years include design changes that have affected production technique, and production practice itself has developed with experience to improve quality and reduce costs to a very marked degree.

The methods and practices described here pertain to pistons characterized by an elliptical skirt having a slot or slots that affect the form of the piston under operating conditions, and the production procedure has developed during the manufacture of several million pistons for automobiles in all price classes, and for a considerable number of marine and industrial engines as well.

As in all highly stressed parts, the first requirement for a satisfactory piston is the proper material analysis which, in the present case, is as follows:

Copper	6.25 to 7.75 per cent
Iron	1.5 per cent, maximum
Magnesium	0.25 to 0.3 per cent
Silicon	5 to 6 per cent
Zinc	0.5 per cent, maximum

[This paper was presented at the Annual Production Meeting of the Society, Detroit, April 23, 1936.]

Piston Casting

The material is received in ingots of 32½ lb. which are melted in small open-hearth, oil-fired furnaces having an exposed discharge well at one end. Two molders dip from one furnace. This type of unit has replaced the rotary-tilting type which required the use of a bull ladle to distribute the molten metal to holding pots at each pair of machines, thereby eliminating one handling of the metal and avoiding a great deal of excess heat in the department. The efficiency and fuel consumption of the newer type are also more satisfactory. The temperature of the metal is closely controlled by automatic means and held at 1250 deg. fahr. Of course, the analysis and physical characteristics of the ingots as received, and of the piston castings, are checked continually, and it is necessary to watch the skimming and cleaning of the furnaces to avoid impurities that otherwise would be very troublesome.

The metal is handled from the melting furnaces to the molding machines with hand ladles, each having a capacity slightly greater than that required for one casting. Permanent molds and permanent collapsing cores are employed; the molds are filled by gravity only, no pressure injection being

used. The molds are heated by a gas flame before production is started after a shutdown to minimize imperfect castings. Molds are split on the vertical centerline of the piston and open and close horizontally; the three-piece cores are withdrawn vertically. The casting is poured with the head up, and a sprue of generous size is desirable to provide for the considerable shrinking of this material. One operator tends two molding machines which are hydraulically operated, and the average output is about 70 castings per hr. per man.

When a casting has cooled sufficiently to be removed from the mold, it is conveyed to the heat-treating, or annealing furnace. The heat-treatment is very important as any variation not only affects the strength and hardness, but also influences the important factor called secondary growth which must be held as constant as possible.

Due to the light weight of the work being handled, an overhead oven has been found practical and conserves a great deal of space. The heat-treating cycle is, of course, controlled by the speed of the conveyor, which speed normally provides for 6 hr. in a heat of 400 deg. fahr. for the analysis of material being considered here.

After heat-treatment the pistons are inspected and unloaded from the conveyor into an inclined bin which feeds the castings within easy reach of a punch-press operator who crops the sprues two at a time. This operation has proved to be fast and economical as compared with the earlier practice of removing sprues and gates with a band-saw.

After cropping the castings are checked for hardness in an automatic testing machine, the specified hardness limit being from 100 to 130 Brinell, although in practice it has been found readily possible to hold to a much closer range. The accuracy of control throughout the foundry operations is evidenced by the fact that castings returned from the machine shop for variation in hardness are practically unknown.

The final operation in the casting department is spot-facing the head of the piston to remove the small projection remaining after the sprue is cropped. The spot-facing also gives a good opportunity for visual inspection of the structure of the casting, as it is in the head of the piston that any defects such as non-metallic inclusions and porosity are most apt to appear.

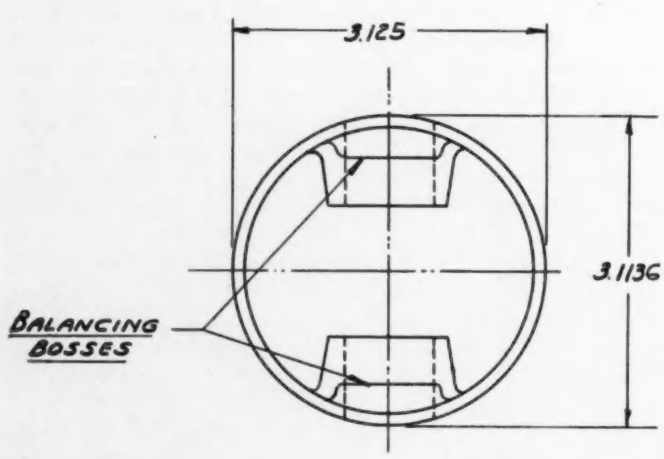


Fig. 1 – Open End of Piston Showing Elliptical Shape and Balancing Bosses

An interesting feature of this department is its compactness, as all operations and a satisfactory storage of aluminum pigs are included in a floor space of only 1200 sq. ft., and an output of 18,000 castings per 24-hr. day can be maintained indefinitely.

From the foundry the castings are shipped to the various motor-building plants in which the technique has become more or less standardized in so far as dimensional tolerances and physical characteristics of the finished piston are concerned.

This type of piston is designed to fit the cylinder bore with a very small clearance, averaging about 0.001 in. in a cold motor, the expansion under operating temperatures being compensated for in the slotted skirt. See Fig. 1. Quiet operation, especially with cold engines, demands extremely close fits and therefore narrow machining tolerances for all bearing dimensions.

The skirt of the piston is oval, the shape being that of an ellipse whose minor axis is 0.011 in. less than its major axis, the major axis being at right angles with the wrist pin. The skirt also is tapered, the mean diameter at the open end of the skirt being 0.0005 in. greater than immediately under the lower ring-grooves. Four ring-grooves are provided, the two top grooves being fitted with compression rings, whereas the two lower grooves are fitted with oil rings. Ten 1/8-in. oil holes are drilled in the No. 3 ring-groove, whereas four 1/8-in. oil holes are drilled in No. 4 ring-groove. These holes are provided to allow the oil collected by the oil rings to return through the holes to the oil pan, thereby preventing oil pumping. There also are two 1/8-in. holes drilled in the No. 4 ring-groove. These holes are approximately 120 deg. apart, and the 3/32-in. horizontal slot previously mentioned terminates in them. Two 7/32-in. holes also are drilled at each end of the vertical slot.

Machining Operations

Machining methods naturally will vary somewhat between motor plants set up for moderate volume and the large-volume plants. These set-ups vary namely in whether or not the piston will be turned and grooved in one or two operations; whether the slots will be milled in one or two operations; whether the elliptical contour will be ground on external grinding machines with a cam attachment, on centerless grinding machines, or diamond-turned on special equipment; whether the piston-pin holes will be finish-reamed or diamond-bored; and whether engineering specifications demand a common weight piston for service in the field and an anodic-plating process for long life.

In one plant the operations are as follows:

Operation No. 1 – Bore, face, and chamfer open end. This operation is performed on an 11-in. engine lathe equipped with a special two-jaw, air-operated chuck which locates on the outside diameter of the piston and, due also to the design of the permanent molds, assures the correct wall thickness. When an internal chucking method was used, we were continually having trouble with the adjustment of plungers in the chucks. The face of the open end is held from the inside of the head end by means of a revolving stop set in relation to the tool bit and mounted in the same tool block. This arrangement allows us to maintain the proper thickness through the head end. Due to the short depth of cut the tools are hand-fed to the work. A production of 300 pieces per hr. per operator is obtained.

Operation No. 2 – Drill, ream, and chamfer piston-pin holes and cut lock-ring groove. This operation is performed on a special two-way horizontal drilling machine equipped with multiple heads, hydraulic feed, and drum-type hand-indexed four-station fixture, each fixture holding two pistons. The part is located in the bore and clamped down against the faced open end. A V-type plunger locating on the wrist-pin bosses insures the equalizing of stock in the boss. Coolant is used on this operation. Production is 500 pieces per hr. per operator.

Operation No. 3 – Rough- and finish-turn outside diameter; rough- and finish-form and chamfer ring-grooves; and rough- and finish-face head end. This operation is performed on a six-spindle automatic chucking machine. This machine is fully automatic, the operator simply loading and unloading the part. In this operation a driving pin is inserted through the wrist-pin bosses and through a driver located between the bosses which draws back locating the piston on the bored diameter and against the face of the open end insuring the correct overall dimension when the head end is faced.

The skirt of the piston is not cam-turned although such a turning attachment could be installed. 0.010 in. of stock is left on the outside diameter for grinding.

The ring-grooving tools are held in magazines which have previously been set up to expedite tool changes. All tools on this operation are tipped with tungsten carbide. Coolant is used. This machine produces 180 pieces per hr.

Operation No. 4 – Drill ten 1/8-in. holes in No. 3 ring-groove. This operation is performed in a special horizontal drilling machine equipped with ten motor-driven spindles mounted on a common base and spaced at the proper angles. The work is placed in a pot-chuck type of fixture with a locator which registers in the No. 1 ring-groove. The spindles are fed forward toward a common center by means of a foot-operated lever. No coolant is used and one operator produces 440 pieces per hr.

Operation No. 5 – Drill five 1/8-in. holes in No. 4 ring-groove and four 7/32-in. slot holes. This operation is practically identical with Operation No. 4.

Operation No. 6 – Drill four 5/32-in. holes through piston-pin bosses. A No. 2 sensitive drill press equipped with a two-spindle multiple head is used. A clamping attachment mounted on the quill is used to operate disappearing pins which locate and clamp the piston in the pin bosses. Production per hr. is 450 pieces.

Operation No. 7 – Saw horizontal and vertical slots. (See Fig. 2.) The machine used on this operation is a special milling machine having one horizontal and one vertical spindle, having separate motor drives but mounted on a common slide which is hydraulically fed to the work. A rotary table is mounted on the bed of the machine. The fixtures on the rotary table are so arranged that, when the milling heads advance, each spindle is cutting a slot in separate pistons. One operator produces 600 pieces per hr.

Operation No. 8 – Rough cam-grind outside diameter. This operation is performed on a centerless grinder, and the method is rather unique.

The regulating wheel has been replaced with a nitrided six-lobe cam having, between its high and low point, the desired difference to give the right elliptical shape to the piston. The machine is arranged with an automatic infeed attachment. After the operator has inserted a pin through the piston-pin hole of the piston and engaged it with the driver,

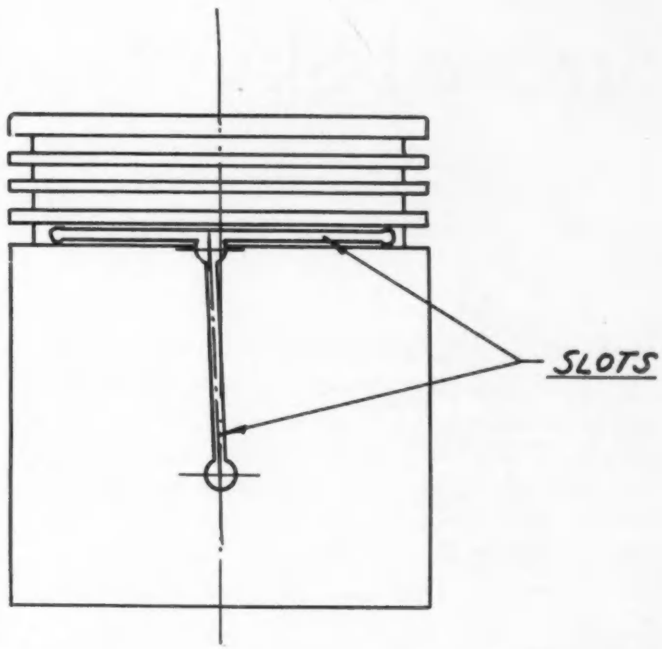


Fig. 2 – Side View of Piston Showing Slots

he presses the starting button which moves the regulating slide forward; simultaneously the work driver begins to turn in synchronism with the regulating-wheel cam so that the piston is ground elliptically in the proper relation to the piston-pin hole. (See Fig. 1.) After the grinding cycle is completed the regulating-wheel slide returns to its starting position and the work driver stops. This arrangement permits the operator to load and unload the work while it is not rotating.

The method of grinding just described is principally a generating method. The shape of the piston is generated by its movement to and from the grinding wheel which movement is influenced by the multiple-lobe regulating-wheel cam pushing it to and from the grinding wheel over the top of the angular top work-rest blade. You will note from this description that the grinding is done entirely by the centerless method, and accuracy of shape and size is obtained due to the exceptionally rigid support of the piston during the grinding operation.

An ellipse having a difference of 0.011 in. between the major and minor diameters is ground by this method removing a maximum of 0.015 in. stock. A production of 300 pieces per hr. is obtained.

Operation No. 9 – Finish cam-grind outside diameter. This set-up is a duplicate of the method used for rough grinding. 0.005 in. of stock is removed in this operation. The piston is ground with a 0.0005-in. taper, the greatest diameter being at the open end. The operators check their work with amplifying gages.

Operation No. 10 – Rough and finish diamond-bore piston-pin holes. This operation is performed on a four-spindle diamond-boring machine having two roughing spindles mounted on one end and two finishing spindles mounted on the opposite end. Two pistons at a time are bored. The fixture is hydraulically fed toward the roughing spindles, then reversed

(*Continued on page* 450)

Stressed-Skin Structures for Aircraft

By Don R. Berlin
Chief Engineer, Curtiss Aeroplane & Motor Co., Inc.

THIS paper includes comments on advantages over other types of structure; adaptability to machine manufacture for production and design considerations to best accomplish normal life of structure which may be expected under continued service; and frequency of overhaul necessary to basic structure during life of airplane.

IT is quite natural for the designer and the factory manager, considering for the first time the use of a stressed-skin structure for a new design of airplane, to assume that such a structure is somewhat complicated and that the cost of design and of manufacture might prove prohibitive. It is perfectly logical that such an assumption should be made, for there have been some very outstanding examples of expensive all-metal airplanes in the past. However, during the past six or seven years there has been an ever-increasing tendency toward all-metal design due in a great extent to the demand of the operators for this type of structure on account of the decreased cost of maintenance and longer life of the structure and, therefore, a great amount of work has been done in the matter of simplifying design and reducing cost of manufacture.

There is no doubt but that, for a small quantity of airplanes to be built, the costs are higher for all-metal structures than for the old steel-tube fuselage, wooden-wing, fabric-covered type. There are several reasons for this higher cost, among which may be listed increased cost of materials, increased tooling required for fabrication of parts, and the more elaborate jigs required for assembly. However, in quantity production if cost of production is kept foremost in the designer's mind, it is the author's belief that all-metal structures, particularly of the stressed-skin type, lend themselves to low-cost production better than any other type. In the design of all parts such as fuselage rings, wings, and tail-surface bulkheads, aileron, elevator and rudder ribs, and all small parts requiring forming, it should be borne in mind that, in order to save in the cost of fabrication, parts should be designed so that a hydraulic press, flanging machine, or drop hammer can be used to advantage in the fabrication of the part.

Any part that requires flanging along a curved surface must, in most cases, be made of annealed stock and heat-treated after forming. When a part is flanged along two adjacent sides, care should be taken to provide sufficient radius at the intersection to prevent cracking in forming and thus causing a rejection of parts after fabrication. An example of what happens when an insufficient radius is provided is indicated in the upper view of Fig. 1, while the lower view indicates the manner in which this cracking can be avoided. In the case of nose ribs where it is desired to form a continuous flange around the leading edge of a small radius, it is necessary to reduce the flange width around the nose section in order to prevent cracking and excessive wrinkling due to shrinking the flange around the radius (see

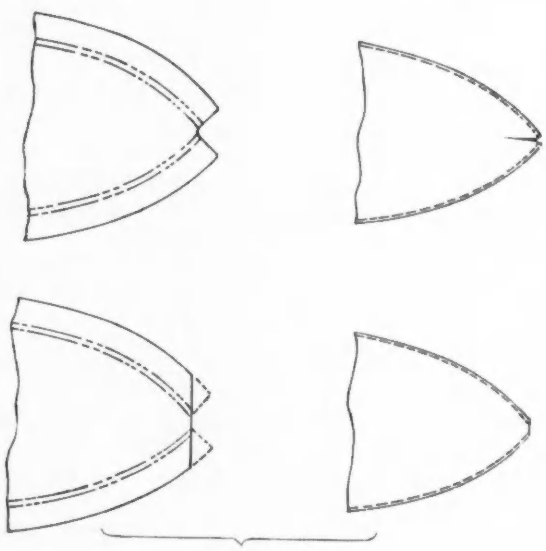


FIG. 1

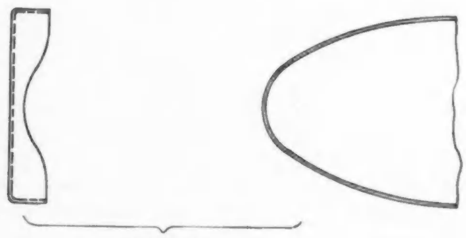


FIG. 2

Fig. 1 (*Above*) – Example of Cracking Caused by Insufficient Radius When Flanging Along Two Adjacent Sides (*Below*) – Manner in Which this Cracking Can Be Avoided

Fig. 2 – Reduction of Flange Width Around Nose Section of Nose Rib To Prevent Cracking and Excessive Wrinkling

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 4, 1936.]

Fig. 2). Flanges should never be bent to more than a 90-deg. angle when the part is to be fabricated over a block in the hydraulic press or on a drop hammer. In addition, care should be taken to insure that ample bend radii are used even for annealed stock in order to prevent cracking along the bend.

Several different types of fuselage structure are in use at the present time, varying from the type in which a skin of sufficient stiffness is used to carry the bending stresses without longitudinal stiffening and in which the skin is wrapped around the fuselage in sections, to the type where a thin skin is employed and assembled in long narrow sheets running lengthwise of the fuselage and having integrally formed stiffeners as indicated in Fig. 3. The first type, although it may be a somewhat simpler construction, results in a heavier structure than that which can be obtained by the use of light stiffeners with thin-gage skin. The type of structure in which the stiffeners are formed integrally with the skin would be ideal if it were possible to form the stiffener on a brake. However, this method would require that the fuselage line be straight in order to assemble the sheet without wrinkling but, as the normal fuselage is curved along any longitudinal line, if this type of stiffener is to be used it must be formed with the sheet held in a block conforming to the fuselage contour with the forming done by hand, which is quite expensive. This type of stiffener has an additional disadvantage in that, due to the spacing of stiffeners required to properly reinforce the skin to carry bending loads, the skin has to be put on in a number of narrow sheets.

In order to simplify the problems of fabrication and assembly, it is advisable to use extruded dural angles (see Fig. 4), generally referred to as bulb angles, and to assemble them to the fuselage frame members, or rings, prior to starting assembly of the skin. The Aluminum Co. of America can now furnish such sections with a minimum thickness of 0.040 in. and, although this construction results in more weight in stiffeners than when integrally formed stiffeners are used, shop costs can be reduced considerably due to the fact that no forming is required either on stiffeners or skin. The fuselage frame member is cut out to permit the use of a continuous stiffener, as may be seen by Figs. 3 and 4 and, in the case of the extruded stiffener, it is necessary to clip it to the frame. This attachment may be done by the use of a separate clip or by forming the clip integrally with the frame member as indicated in the left-hand view of Fig. 4. If the fuselage frame members are to be formed over a block in the hydraulic press having a channel section as indicated in Fig. 5, this clip may be formed in the same operation, thus eliminating two rivets at each stringer-frame joint. A Z-type frame member, as indicated in Fig. 6, has some advantage over the channel-type section, provided that it is made on a flanging machine, as the inner flange does not interfere with the riveting of the skin to the frame.

Another type of longitudinal-stiffener and frame-member combination occasionally used is indicated in Fig. 7. In this combination the longitudinal stiffener only is attached to the skin and the frame member is formed to fit the lower flange of the stringer. This construction results in considerably less riveting over the condition where the ring is brought out to the fuselage skin line as shown in Figs. 3 and 4, but it has some disadvantage in that torsional stiffness of the fuselage will be somewhat reduced due to non-support of the skin at the frame sections.

Wings of stressed-skin structure consist of a series of shear

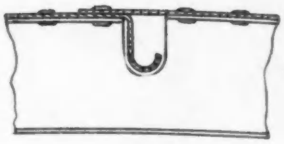


FIG.3

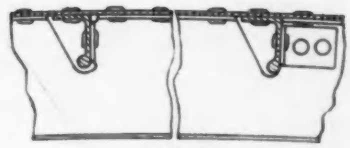


FIG.4

FIG.5

FIG.6

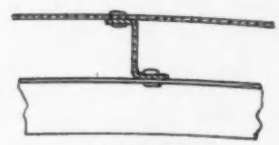


FIG.7

Fig. 3 – Integrally Formed Longitudinal Stiffener

Fig. 4 – Extruded Longitudinal Stiffeners Clipped to Fuselage Frame

Figs. 5 and 6 – Fuselage Frame Members

Fig. 7 – Longitudinal Fuselage Stiffener Riveted to Skin with Fuselage Frame Member Attached to Stiffener Only

webs, bulkheads (or ribs) spaced at approximately 2-ft. intervals along the span and a covering of skin properly reinforced by stiffeners to carry bending stresses. The normal type of shear web consists of a flat sheet having assembled to it extruded angles, or bulb sections, for attachment to the skin at the top and bottom. These webs normally have flanged lightening holes located at intervals between bulkhead stations in order to provide access holes for bucking rivets in the flanges of the bulkheads at the time of wing assembly. Another type of web which has been used to some extent is that which employs tension diagonals (Fig. 8). This type of web, however, is not used extensively due to the expense involved in fabrication as, in the normal tapered wing, a standard die can not be used for cutting out the four triangular sections necessary to form the diagonals. Aside from this design, a plain flat sheet without cutouts provides the best shear structure, and lightening holes, as shown in Fig. 9, are advisable only from the standpoint of providing access holes for riveting.

A diagrammatic sketch of a cross-section of a wing of this type is given in Fig. 10 and indicates several different types of wing stiffeners and their attachment to the bulkheads. All of the types of stiffeners now in use are not shown in this sketch, but it is believed that sufficient types are indicated to permit an outline of the advantages and disadvantages of the various types. Stiffeners indicated at *A* and *B* are extruded sections, the stiffener at *A* being a Z-section and the one at *B* a bulb angle. In each case the stiffener is attached to the wing bulkhead by a clip in order to provide sufficient support to the stiffener to prevent buckling of the free edge under compression load. The bulb section indicated at *B* provides a much easier assembly, but the free edge of this stiffener will start buckling under compression load at a lower stress than the stiffener shown at *A*.

In each of these cases the bulkhead is cut out for the stiffener and has a flange which rivets to the skin between stiffeners. At *C* a bulb-angle stiffener is also shown similar to *B* but, in this case, the bulkhead does not come down to the skin line but is attached to the stringers by clips. This method of attachment of the bulkhead to the stringer is used quite frequently and eliminates cutouts in the bulkhead and additional riveting to the skin but again, as in the case of the fuselage ring member which attaches to the fuselage stiffener

only (Fig. 7), torsional stiffness of the wing is reduced by non-attachment of the bulkhead flange to the wing skin.

At *D* is shown a closed-section stiffener in which the two flanges are attached to the skin. This type of stiffener has some advantages in that it can be tapered throughout its length, thus saving some weight and, in addition, it does not require any attachment to wing bulkheads. However, this type of stiffener offers very serious assembly difficulties as the flanges must be formed to fit the wing contour and, as the curvature of the wing at *D* is differenι from that at *A* or *B*, the use of this same type of stiffener at either *A* or *B* would require that the flanges of the stiffener be bent at a different angle to fit each of these sections. If this bending is not done, the section of skin between the two flanges will be pulled to form a flat spot in the wing contour if the flanges are not bent down enough, and will buckle the skin outward if the flanges are bent down too much. From this explanation it may be seen that the use of this type of stiffener requires considerable amount of work in the process of fabrication as stiffeners with flanges at several different angles must be kept in stock and so marked that they can be selected for use in the proper location. On the other hand, the extruded section, which has a single flange attaching it to the skin, can be assembled to the skin without fear of distortion of the contour, the only care required being in attachment of the clips tying it to the bulkheads to insure that the stiffener is not pushed away from its natural position.

In considering the process of assembly of a wing of this type to provide the best arrangement for production assembly for the shop, we will consider the structure indicated in this diagram (Fig. 10) back to and including Web No. 2. The structure will be broken down into bench assemblies as follows:

(1) Nose-section assembly in which the stiffeners and nose bulkheads are assembled on to the nose-cover sheet in a separate assembly jig.

(2) Web. No. 1 assembly wherein the extruded flanges and all intermediate stiffeners, and so on, are assembled on to the web in a separate jig.

(3) Assembly of stiffeners on to the top skin between Webs Nos. 1 and 2.

(4) Assembly of stiffeners on to the lower skin between Webs Nos. 1 and 2.

(5) Assembly of extruded flanges and intermediate stiffeners on to Web No. 2.

This process continues for the remainder of the wing and, after all parts are completed, they go to a main-assembly jig in which the nose section is first placed and No. 1 Web located. Then follows location of bulkheads from Web No. 1 to Web No. 2 and assembly of the top and bottom skins on to which stiffeners already have been assembled. The attachment of stiffeners to bulkheads and bulkheads to skin between Webs Nos. 1 and 2 is then accomplished, and No. 2 Web then put into place, the process continuing in this manner for the remainder of the wing. For convenience of explanation only two webs have been indicated in this sketch (Fig. 10).

This type of wing structure is basically quite simple from a standpoint of structural analysis and design and lends itself quite readily to low cost in production if the designer is familiar with shop processes and methods of fabrication and assembly and if, in addition, he consults freely with factory personnel in the matter of detailed design for ease in fabrication and assembly. Probably the most expensive feature of this type of structure is the matter of riveting and it is, therefore, essential that all assemblies be laid out in such a manner as to require a minimum time for putting in the required rivets. Until recently in the use of duralumin rivets, it has been necessary to heat-treat the day's run of rivets and store them in small refrigerators in the various departments in the shop where riveted assemblies are made, the rivets being taken out as needed. Rivets heat-treated and placed in refrigerators in this manner can be used within a period of approximately 24 hr. as the reduced temperature delays the aging process after heat-treatment. However, once rivets are

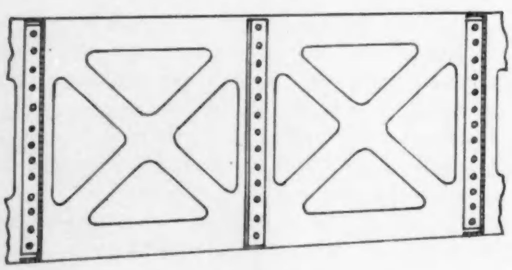


FIG. 8

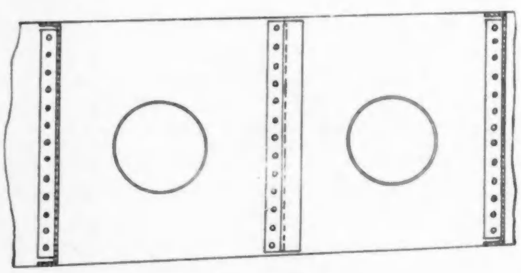


FIG. 9

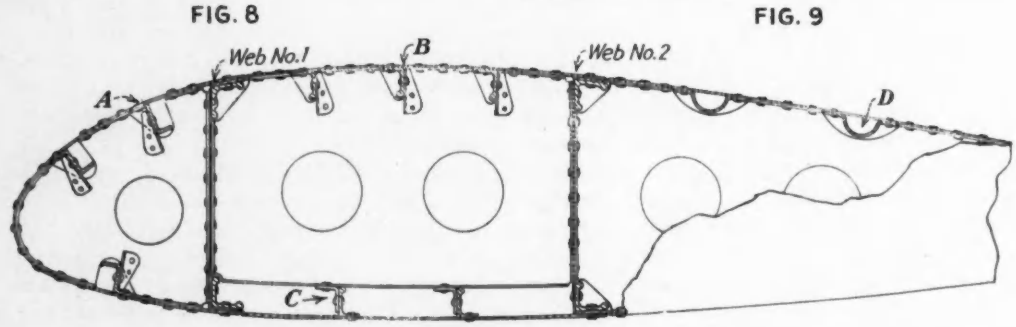


FIG. 10

Fig. 8 – Shear Web Employing Tension Diagonals

Fig. 9 – Shear Web with Flanged Lightening Holes

Fig. 10 – Cross-section of Wing Structure Showing Several Types of Wing Stiffeners

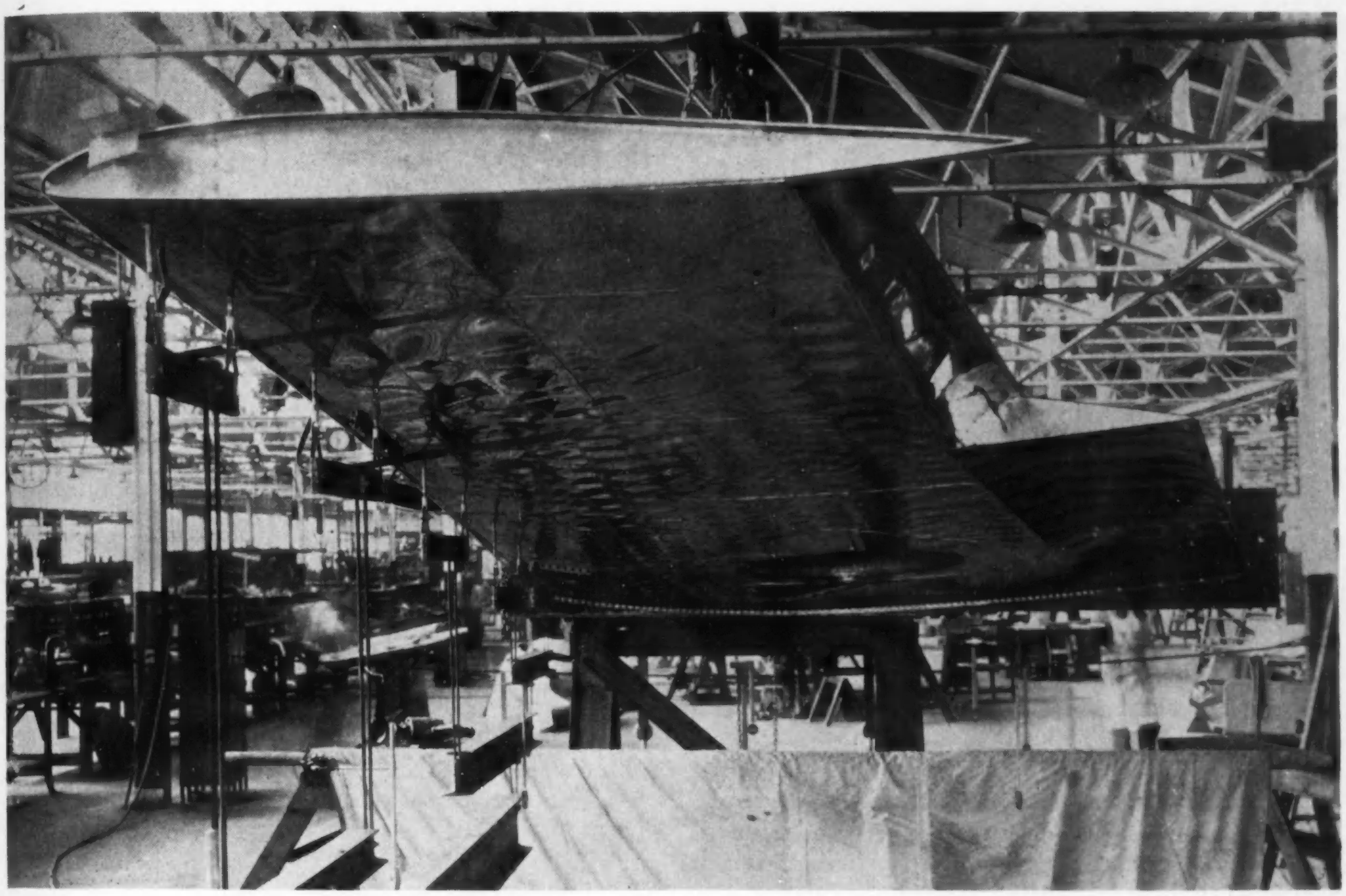


Fig. 11 – Wing of Stressed-Skin Structure under Full Inverted-Flight Load

taken out of the refrigerator they must be used within a period of ½ hr. as, after this period, they have aged sufficiently so that cracking occurs when the rivet is headed-over. Recently the Aluminum Co. of America has developed a rivet, designated as A17S, which is in the semi-heat-treated state and can be used without further heat-treatment, thus eliminating the care and additional work formerly required in the handling of rivets.

There is still the problem of the time involved in the location of rivet holes, in insuring that all drill chips have been removed from between sheets or sheets and stiffeners, and in actually driving or squeezing rivets into place. Rivet machines that have been developed for various types of rivets, for riveting small assemblies, or even for riveting stringers on to large sheets, save a considerable amount of time. However, there is still much to be hoped for in this respect, particularly now that flush-riveting has been found to be so important from a standpoint of reduction of air resistance.

A great amount of work has been done in the past three or four years in experimental work on electric spotwelding of aluminum alloys in an attempt to eliminate a great amount of the expense now involved in the riveting process. To date electric spotwelding of aluminum alloys is not acceptable for basic structures but is being used to quite an extent in the assembly of non-structural parts. In addition to the elimination of a large portion of the time now required for riveting, electric spotwelding holds forth the additional advantage of providing a smooth surface which now can be obtained only by the use of flush rivets.

One of the great advantages of stressed-skin structures is that the stresses are quite uniformly distributed throughout the structure. An interesting example of the distribution of stresses throughout a wing of this type can be seen in Fig. 11. This is a photograph of a wing under full inverted-flight load, the total load being applied at approximately 8 per cent of the chord aft of the leading edge of the wing and distributed along the span so as to represent the application of an air load. This particular wing has a large cutout in the lower skin near the root of the wing and this photograph gives a good indication of the manner in which the structure is carrying the load, that is, whether one section is highly stressed while another carries but a very small portion of the load. It will be seen by inspection of this photograph that, although the load is applied at the leading edge of the wing, the ripples in the skin are quite uniformly distributed over the entire lower surface of the wing up to the wheel cutout and are carried around this cutout in such a manner that the stresses in the portion of the wing inboard and aft of the cutout seem to be approximately the same as those inboard and forward of the cutout as indicated by the intensity of the ripples. These forces are distributed in this manner due to the fact that the wing is in reality an assembly of small box sections, each complete in itself, therefore forming a so-called multicellular structure which is torsionally very rigid.

The fact that stresses are distributed so uniformly permits the use of a continuous flange at the root of the wing for attaching the wings together. The same thing can be done with any section of a fuselage with the exception that the

(*Continued on page* 458)

Sapphire and Other New Combustion-Chamber Window Materials

By George Calingaert, S. D. Heron, and Ralph Stair *

A STUDY has been made of materials available and suitable for the construction of transparent windows employed for observation inside the combustion-chamber. Materials that can be used include quartz, spinel, sapphire, periclase, and fluorite.

The transparency of these materials in the infra-red region increases in the order just mentioned. Mechanical strength varies greatly, being highest for sapphire and lowest for fluorite.

Resistance to the chemical action of the combustion products varies widely. Sapphire is not attacked by any products of combustion, including the lead oxide present when leaded fuels are used.

The choice of a suitable material for a given investigation will be based on a judicious balance of the four chief characteristics: mechanical resistance, chemical resistance, transparency, and cost.

EVER since the automotive engineer has become aware of the importance in engine operation of the chemical reactions that take place in the combustion-chamber, numerous devices have been designed to permit observation of the inside of the combustion-chamber during operation of the engine. Not satisfied to measure temperature, pressure, and so on, investigators have used optical methods of observation, either visual or involving measurement and even recording of radiation intensity, wave length and so on. Of all the window materials available, quartz has been used almost exclusively heretofore.

This paper presents data on the characteristics and use of several other available window materials.

Window Construction. – The various materials were cut and ground in the shape of cylinders, usually around 5/8 in. in diameter and 5/16 in. thick, with beveled edges. The two flat surfaces were polished, whereas the others were only ground to a fairly smooth finish. The mounting of these windows is illustrated in Figs. 1 and 2. Inspection under polarized light and comparison with windows made of flat-disc shapes indicated that the use of the beveled joint with the soft-aluminum gasket gave a perfect seal without setting up any of the strains that are evident in the flat-disc type of window.

Window Mounting. – The window mountings shown in Figs. 1 and 2 are both adapted for threading into standard 7/8-in. diameter 18-thread spark-plug holes. The type shown in Fig. 1 has a 5/8-in. diameter clear aperture, and that shown in Fig. 2 has a 1/2-in. diameter clear aperture. In both designs the body (1), gland nut (2), and thrust ring (3) are made of S.A.E. 1112 free machining steel, the gaskets (4) are made of annealed 99.5 per cent aluminum. The windows (5) that have been used with mountings of the two types are: sapphire, spinel, and periclase.

Properties of Materials. – Engine windows of synthetic fused quartz, synthetic spinel, synthetic sapphire (white), synthetic periclase, and fluorite have been made and tested. Their light-transmission characteristics have been determined and are illustrated in Fig. 3 over the range 0.2 to 15 μ ($\mu = 0.001$ mm.).

The apparatus used in determining the ultra-violet transmission consisted of a quartz-fluorite achromatic-lens spectrometer, quartz prism, portable vacuum thermopile, and iron-clad Thomson galvanometer. The source of radiation was a vertical "Uviarc" quartz mercury-arc lamp.

The spectroradiometer for determining the transmission in the infra-red consisted of a mirror spectrometer, fluorite prism, portable vacuum thermopile, and iron-clad Thomson galvanometer. The source was a Nernst glower.

Both spectroradiometers are part of the equipment of the radiometric laboratory of the National Bureau of Standards and are described in publications of that institution.

The transmissions given in Fig. 3 all refer to samples 5.45 mm. thick.

The most significant properties of the materials are given in Table I, and are discussed further in following paragraphs.

Quartz. – Fused quartz is available in large sizes. Its only limitations are (a) its attack by PbO from leaded fuels, (b) its low transmission in the infra-red, and (c) its poor me-

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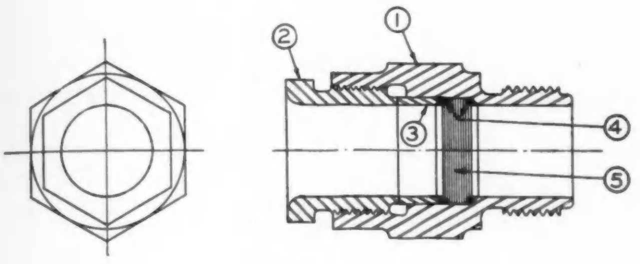

Fig. 1

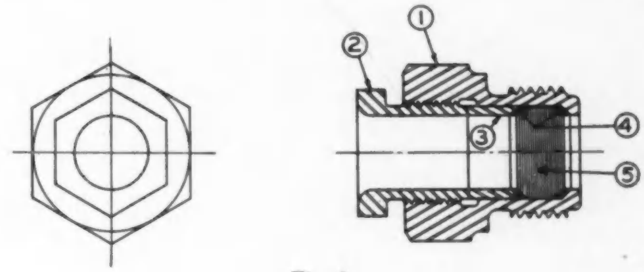

Fig. 2

Figs. 1 and 2 – Mountings for Combustion-Chamber Windows – 5/8-In. Aperture, Fig. 1; 1/2-In. Aperture, Fig. 2

chanical strength. For use where these three factors are unimportant, it still remains the best material.

Spinel. – Spinel closely resembles sapphire in its properties, except for the absorption band shown by the specimen tested. The behavior of two windows subjected to engine tests has been very satisfactory in all respects.

Sapphire. – Synthetic white sapphire is more expensive than quartz and is not available in large sizes. For small windows

Table I—Properties of Engine-Window Materials

Material	Chemical Composition	Crystalline Form	Cleavage	Mechanical Strength	Chemical Resistance to PbO	Chemical Resistance to CO_2 and H_2O	Limit of Transparency (30 Per Cent), μ
Quartz	SiO_2	Amorphous	......	Poor	Poor	Good	3.9
Spinel	$MgAl_2O_4$	Amorphous	......	Good	Good	Good	5.4*
Sapphire	Al_2O_3	Amorphous	......	Good	Good	Good	5.9
Periclase	MgO	Cubic	(001), moderate	Fair	Good	Fair	7.8
Fluorite	CaF_2	Cubic	(111), pronounced	Poor	Good	Good	10.4
Rock Salt	NaCl	Cubic	(001), moderate	Poor	Good	Very Poor	18

*The specimen tested had a greenish cast, which appearance suggests that the sharp absorption band observed at 3 μ may be due to some impurity.

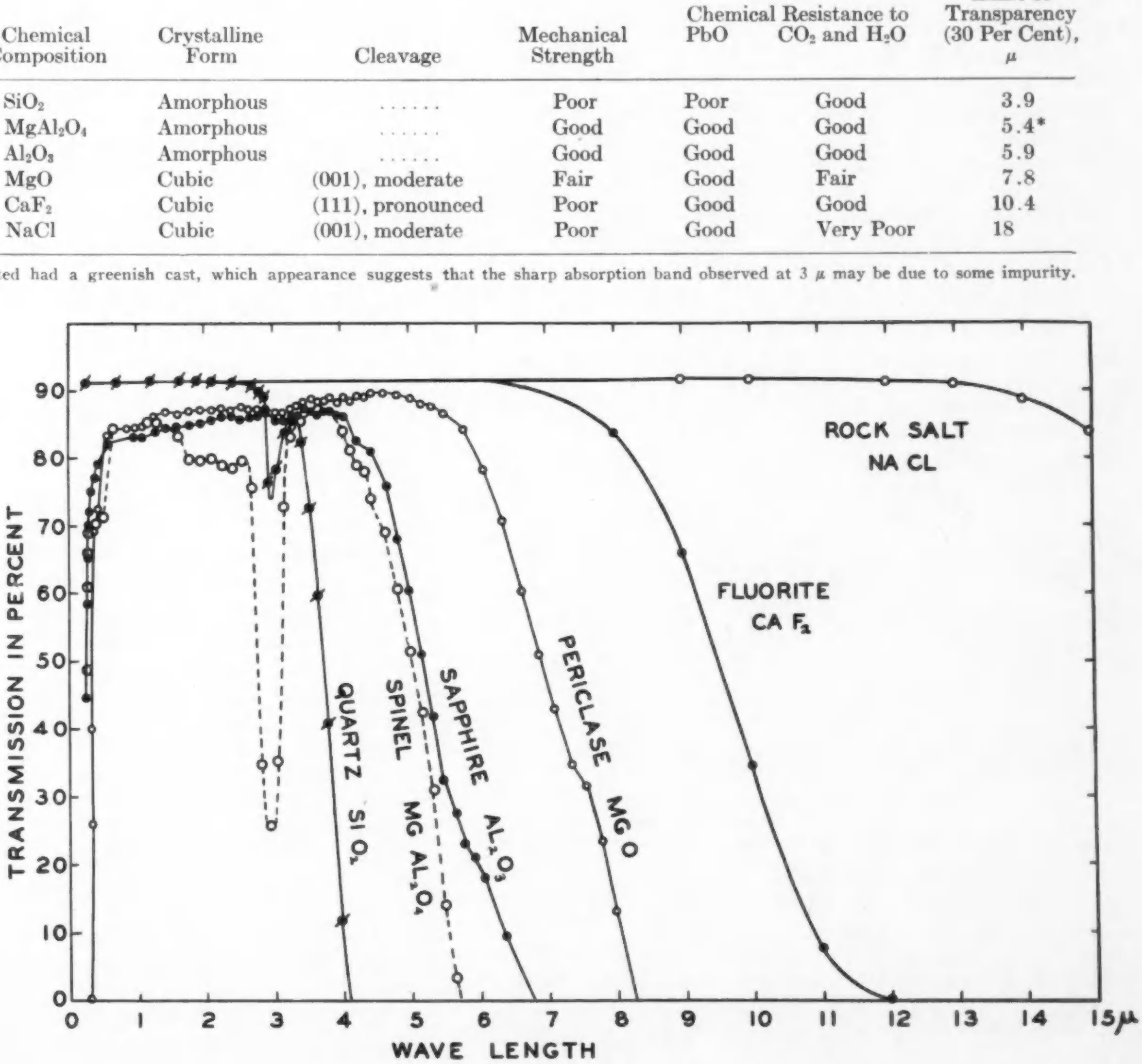

Fig. 3 – Light-Transmission Characteristics of Combustion - Chamber Window Materials

(up to 3/4 in. in diameter), it has the following advantages over quartz: (a) greater mechanical strength, (b) higher infra-red transmission, (c) higher coefficient of expansion, and (d) greater chemical resistance.

When cost is not a primary factor, sapphire appears to be one of the most interesting materials. Inasmuch as it is made synthetically, it may become possible to obtain it in large sizes if desired.

Two windows of 5/8-in. outside diameter and 1/2-in. diameter clear aperture have been engine-tested over considerable periods in a variety of engines. In some cases engine conditions that have been severe enough to cause fluxing or breakage of quartz windows have been entirely without effect on the sapphire windows. The sapphire windows have shown that they will stand very rough treatment in respect to frequent mounting and demounting and the use of very heavy clamping pressures. Removal of lead and other deposits is readily accomplished with a wet rag and pumice soap.

Periclase. – Periclase is now obtained as a byproduct in the preparation of fused MgO.[1] Its rather high transmission in the infra-red makes it a valuable addition to the list of materials suitable for engine windows. One window has been engine-tested for 5 hr. with entirely satisfactory results and without sign of attack under fairly severe conditions in a hot-running engine.

Fluorite. – This material has been used in the construction of engine windows. Its pronounced cleavage and its poor resistance to heat shock make it much less satisfactory mechanically than any of the other materials described. It can be used only where high transparency in the infra-red is of primary importance. Also, in order to obtain the maximum mechanical strength, the flat surfaces should be cut parallel to the (111) cleavage plane.

Rock Salt. – This material is included here only for the purpose of comparison. Its high transmission makes it very valuable in the construction of optical systems. It lacks, however, completely the mechanical and chemical characteristics of a good engine-window material.

Conclusions

The data given in this paper should be of value in guiding the choice of material for an engine window. By appropriate choice of materials it may now become possible to study selectively the emission in various regions of the spectrum.

It may also be that samples of spinel and periclase can be found with selective absorption bands, thus increasing the selectiveness of the transmission studied.

Also, the combination of two or more layers of different materials may lead to a still greater mechanical and chemical resistance without decrease in transmission.

[1] See the *Journal of the Optical Society of America*, Vol. 25, July, 1935, pp. 207-210; "Optical Properties of Magnesium", by John Strong and R. T. Brice.

Manufacture of Elliptical-Skirted Pistons

(Continued from page 443)

and fed toward the finishing spindles. The piston is located in a pot-chuck type of fixture locating on the outside diameter and with the head end down resting on fixed buttons. A top plate carrying equalizing vees which contact in the piston-pin bosses, serves both to locate and to clamp the piston. Tungsten-carbide tools are used for roughing and diamond tools for finishing; 0.010 in. of stock is removed by the finishing tool, and a limit of 0.0002 in. in diameter is held. Coolant is used on this operation. The production per machine is 150 pieces per hr., one operator running two machines.

Operation No. 11 – Weigh and machine to common weight. This machine consists of a cam-feed drilling unit mounted vertically on a base and set in relation to a weighing scale with a special platen mounted directly above. To avoid distortion in clamping, the piston is placed in a sleeve with a pin inserted through the wrist-pin bosses and sleeve. The piston, pin, and sleeve are placed on the scale platen and, after coming to rest, the sleeve is clamped. A heavy piston naturally will come to rest closer to the cutter than a lighter one. The cutter, always rising to a known height, insures each piston to become the same weight. A limit of plus or minus 2 gm. is held. A production of 200 pistons per hr. is obtained.

Operation No. 12 – Inspect ring-grooves and piston-pin hole. Ring-grooves are inspected for width and squareness. The squareness gages are in the shape of a half-circle and are revolved in the groove.

An electrolimit gage is used on the piston-pin hole, which checks the size to 0.0004 in., the allowable limit on the hole.

Operation No. 13 – Anodize. All pistons are given an anodic treatment that reduces wear to a minimum. This treatment is essentially that of making the piston the anode in an electrolytic bath of sulphuric-acid solution. Pistons are supported on an aluminum rack while undergoing this treatment, each rack containing ten pistons.

Racks of pistons are loaded on a conveyor chain which moves through a series of tanks for the following operations: dip in cleaning tank the solution of which is heated to 160 to 180 deg. fahr.; cold-water rinse; dip in anodizing tank of 25 per cent sulphuric-acid solution at 72 deg. fahr.; cold-water rinse; and hot-water rinse.

One notable feature of this process is that, as the coating is formed, the resistance to electrical current increases. The current consumption is 20 amp. per piston (average) and current density is 14 volts.

Summing up, we may say that anodizing is just the opposite from electroplating, and that the piston increases its growth uniformly about 0.0003 in. A production of 700 pistons per hr. is obtained.

Operation No. 14 – Inspect outside diameter, grade for size, and stamp trademark and size. All pistons are inspected for diameter after anodic plating. This inspection is made necessary by the growth of the piston in the plating process. The growth averages 0.0003 in. It appears to be constant on the diameter, in the ring grooves, and in the piston-pin holes.

The pistons are inspected at a temperature of 70 deg. fahr. with electrolimit gages. Four grade sizes are allowed and designated *A, B, C,* and *D* with a variation of 0.0005 in. between grades.

For the temperature control of 70 deg. fahr. we use a fully enclosed tank, 8 in. square and 5 ft. long. Through this tank a brass tube with a 3/32-in. wall thickness is inserted, projecting out at each end for loading purposes. This tube in the tank is entirely surrounded by water. The temperature of the water is controlled by means of a three-way automatic mixing valve. The piston is conveyed through the tube by means of a small chain with driving lugs. Pistons remain in the tube 1½ min. Pistons that are approximately 125 deg. fahr. when placed in the tube are 70 deg. fahr. when removed from the opposite end.

The Chassis Frame - Its Functions and Means for Increasing Efficiency

By D. W. Sherman
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ALONG with larger tires, independent wheel suspensions, higher speeds, and the dynamic and vibrational problems associated with these innovations, came the need for chassis frames having a high resistance to torsional movements. The X-member-type frame has been the most generally adopted means for obtaining increases in this direction.

However, the past few years have seen the need for torsional rigidity in the chassis frame to be intensified. Although considerable gains have been made, in general, these gains have been accomplished not by major improvements in the design of the structure but by the addition of material. Consequently, the weight of the chassis frame has become a serious problem, so much so that in many cases special heavy frames are being used for the open-body types where the need for a stiff frame is acute.

The purpose of this paper is to present certain experimental data pointing the way toward greatly increased chassis-frame stiffness without the weight penalty heretofore associated with such increases. As an example:

In a standard 1935 car, a closed model having a steel top and a trunk, the torsional-resistance value approximated 10,000 ft. lb. torque to twist the structure through an angularity of 1 deg. The chassis frame's resistance alone (the frame being a conventional X-member-type weighing 265 lb.) equaled 1700 ft. lb. For the same weight it has been possible to replace the standard frame with one having, by itself, a torsional resistance equivalent to the stiffness of the complete standard car.

THE particular research program covered by this paper was started originally with one purpose in mind; namely, the development of a frame experimentally to have many times the rigidity of any previous frame. It was felt that information derived from a study of the performance characteristics of a car having such a frame might be valuable in determining the proper trend of future designs.

Neither weight nor costliness of design was of any particular importance in this experimental set-up as there was no thought that a frame running to such extremes in rigidity would ever be produced in quantities – the desire was to produce as stiff a structure as possible within the space available. Although weight and cost were not determining factors in the ultimate design of the frame, it was felt that the design should be of as efficient a nature as possible and, therefore, a good deal of preliminary work and study was put in before the final type of structure was decided upon. This preliminary study led to the following conclusions:

Structure Types

In a structure such as an automobile frame, two basic types of construction are available, each type being of such a nature that torsional movements are resisted inherently. One of these types is the construction wherein the individual members are themselves resistant to torsional or twisting forces. The tubular frame falls in this class. The other is the type in which the individual members are proportioned purely as beams capable of resisting vertical forces, these beams being so disposed in relation to each other that twisting of the frame causes vertical bending stresses to be set up.

If the only quality required in the frame members were torsional resistance, a structure made of circular tubes would result in the greatest ratio of rigidity to weight. All of the material would function directly in shear and with maximum possible efficiency. However, this condition of pure torsion does not exist. The structure is subjected also to vertical forces and, for loads of this nature, the circular section is not proportioned properly. Therefore, it is evident that, regardless of the style of frame used, its members must withstand vertical forces; and, as a circular member is not shaped properly for bending and as any departure from the circular shape to effect a compromise results in a great loss in torsional resistance, the so-called tubular frame does not compare favorably with the type hereinafter described. The reason for

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its unsuitability, when all is said and done, is that a section can be proportioned for handling only one-directional forces efficiently.

The "X"-Member-Type Frame

Fundamentally, the X-member frame is a system of beams, the beams being proportioned to resist vertical deflection. In Fig. 1 is shown a structure consisting of an X-member and a pair of side rails, this structure being supported at three corners and having a vertical force applied at the free corner.

Loaded in this manner, the frame is subjected to torsional forces that tend to disturb its alignment in the horizontal plane. Under these conditions each of the two members comprising the "X" reacts as a beam supported at the center and loaded with a force at each end. The end forces are identical in magnitude and their sum, of course, is equal to the reaction occurring at the center support. Each member furnishes the center support for the other, so that the forces it carries are equal and opposite.

By virtue of this balanced-force condition the four outer ends of the X-member tend to remain in a plane, the degree to which they do so depending upon the vertical stiffness of the X-member arms. For purposes of analysis in connection with the rest of the system they are considered as "fixed points".

The side rails constitute beams carrying loads at each end and supported at two points by the X-member, their vertical stiffness controlling the degree to which deflection takes place. Although in actual practice the load applications are at various points, depending upon location of body mountings and so on, the net result as far as torsional stiffness of the frame is concerned, resolves into the simple force system shown in Fig. 1. Furthermore, although the structure shown is a pure rectangle so as to simplify the illustration, the balanced-force condition of the X-member holds true in cases where the structure is wider at one end than at the other and, in consequence, where the rear arms of the X-member may be longer than the forward arms.

The X-member and the side rails constitute the load-carrying frame members as far as car rigidity is concerned, cross-members performing detail functions such as the support of the motor, gas tank, and so on, and not being depended upon for any major contribution to the frame's torsional rigidity. It has been shown that these main members are capable of resisting twisting of the frame purely by virtue of their rigidity as beams. Therefore, although torsional movement takes place in these particular members upon distortion of the frame, this movement is purely incidental and is in direct proportion to the efficiency of the members as beams. In other words, if the beams were infinitely rigid, no frame distortion would occur and, in consequence, no torsional movement would take place. The members, therefore, need not be resistant to torsion and can be proportioned purely as beams.

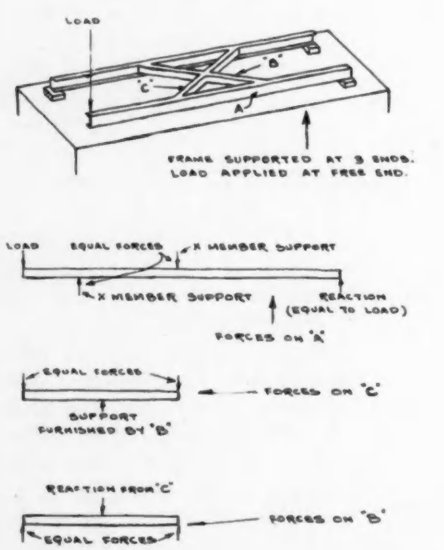

Fig. 1 – Balanced-Force System of X-Type Frame When Subjected to Torsional Loads

As a section can be shaped efficiently for withstanding one-directional forces, but not more than one, the X-member-type frame should be capable of contributing a much greater degree of rigidity to the car structure for a given weight than the type of frame composed of members of tubular section. The former construction was therefore selected for the experimental frame. Furthermore, although rectangular sections having resistance to torsion were used in certain features of the design, it was not done primarily to increase the frame's overall torsional rigidity. The theory of using sections designed for resisting one-directional forces only was rigidly adhered to.

Weaknesses in Conventional X-Member Frames

Both tubular- and X-member-type frames had been used commercially and, although the X-member frame in general developed higher rigidity for its weight than the tubular frame; if the foregoing analysis was correct, the differential should have been greater. A study was made, therefore, to determine the cause of this apparent discrepancy by means of wiggle tests upon various frames, and observations and analysis of movements taking place. As a result of this study it was decided that two conditions were primarily responsible for the discrepancy. These conditions can be listed as follows:

First. – The X-members were so inefficient for carrying beam loads that they did not truly function in this manner.

Second. – The material was not distributed properly throughout the various frame members for obtaining maximum efficiency.

The Conventional X-Member

In Figs. 2, 2A, and 2B are shown three typical X-members, all three having certain fundamental weaknesses resulting in poor carrying capacity. These conditions, or combinations of these conditions, are present to a greater or less degree in all conventional X-members.

In Fig. 2 the rivets attaching the center plates to the four beams are offset. Under the influence of tensional and compressive loads in the flanges of the beams, the center plates tend to crank around their center. Although this action is resisted by the flanges, their resistance is not very great and, in consequence, cranking does take place with beam deflection resulting.

In Fig. 2A the upper and lower center plates are curved for propeller-shaft clearance. The tensional and compressive stiffness of the plates is virtually ruined by a condition of this sort, the plates having little resistance to bending.

In Fig. 2B the beams are offset and tend to keel over as indicated when loaded. The plates attempt to resist this action but, because of their thinness, the poorness of their

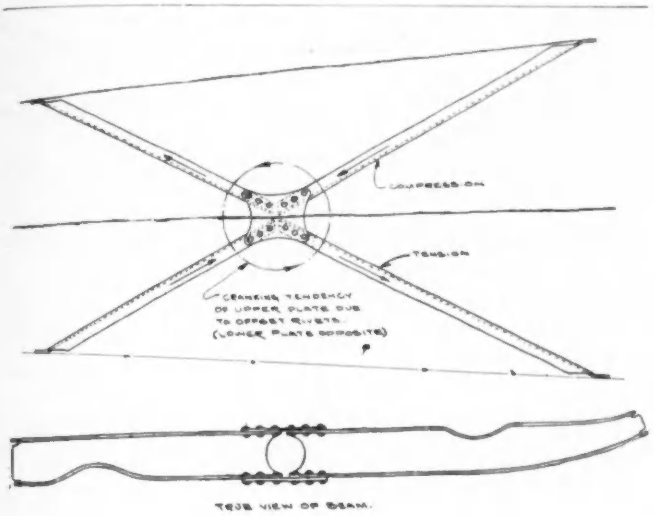

Fig. 2

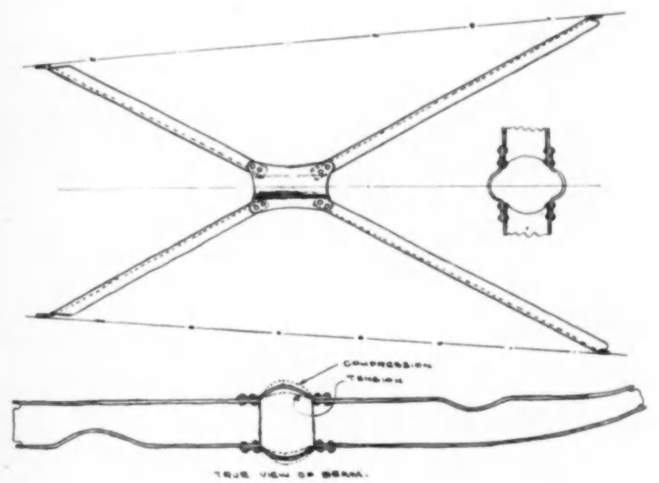

Fig. 2A

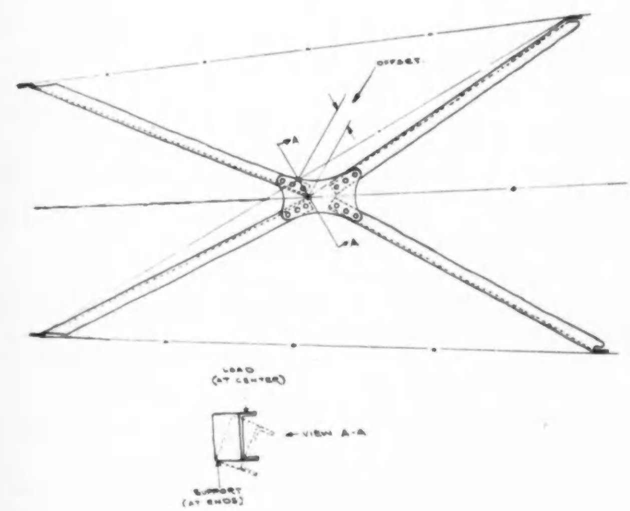

Fig. 2B

Figs. 2, 2A and 2B – Three Typical X-Members Having Fundamental Weaknesses Resulting in Poor Carrying Capacity

connections to the beams, and the large unsupported center area, buckling and distortion takes place and the keeling-over action is not restrained properly.

It can be said, therefore, that a major reason for the weakness of the conventional X-member has been the impossibility of getting a really good center-connection design. Large heavy plates with a great number of connections will improve the condition. However, a heavy weight penalty results; furthermore, in many cases, clearance conditions will not permit of their use.

The channel used for the X-member arms has certain characteristics that render it particularly unsuitable if high rigidity is desired along with lightness. Aside from the fact that it is naturally unbalanced and cannot always be braced wherever necessary, the weaknesses in the center joint of the X-members can be traced back, in most cases, as resulting from the channel's lack of symmetry and from the consequent problems involved in creating a suitable, balanced connection. Because of the need for lightness, the channels have been made from light material – in many cases 3/32 in. – with the result that the flanges, when loaded in compression, react as slender columns with insufficient support, incapable of maintaining their straightness and shirking their responsibilities due to this buckling action. Furthermore, referring to the member shown in Fig. 3, it will be noted that the flanges are not laid out as straight lines, but are curved to clear exhaust pipes, and so on. Where this curved section exists, the buckling action is greatly aggravated; so much so, as shown up by the wiggle tests, that this portion of the flange can be likened to a crooked string, incapable of carrying either tensional or compressive loads.

Summing up, therefore, it seemed that, from practically every angle, the standard X-member was a very poor structure, and that most of its poor qualities could be traced to the use of the channel.

Material Distribution

Earlier, it was stated that the material was not distributed properly throughout the various frame members for maximum efficiency. Briefly, it was found that, in general, the side rails were made from a heavier section than was the X-member and, in some cases, their section was deeper. Considering the side rails and X-member as a system of beams as shown in Fig. 1, it is evident that the X-member carries the highest bending moments. Consequently, the material should be distributed just the reverse of standard practice as far as torsional stiffness is concerned.

A Different Section for the X-Member Arms

In view of the poor qualities of the channel section it was felt that it might be possible to obtain very substantial gains in rigidity in an experimental frame, without the weight increase originally anticipated, if a different section were to be incorporated into the X-member arms. Viewing the X-member from all angles and taking into consideration the buckling action in the standard frame and the problems involved in obtaining a good center connection for the four arms, it seemed as though a section having an "I" shape, rather than the conventional channel shape, should be almost ideal. It was decided, therefore, to adopt this section for the experimental frame, the I-sections to be built up from strip-stock flanges and a flat web by means of arc-welding. (Fig. 4 is an illustration of the X-member as used.) Bearing in mind the greater magnitude of the bending moments in the X-member, it was decided not only to substitute the

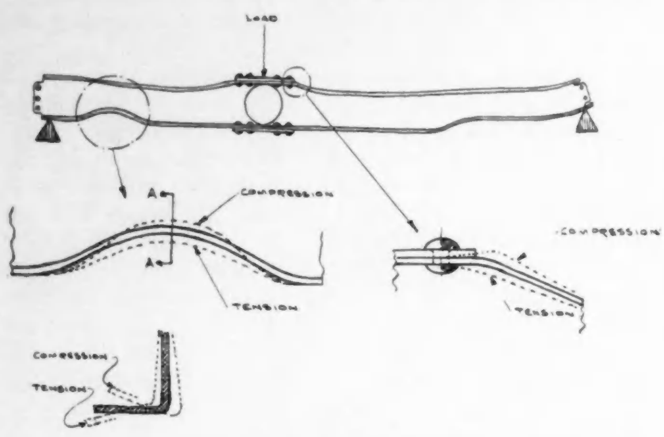

Fig. 3 – X-Member with Flanges Curved

I-section for the channel, but also to concentrate the frame material in this member. For purposes of maximum stiffness, the X-member was made as deep as road clearance permitted which, in this particular case, allowed the use of a member 10-in. deep at the center. Except for a minor reduction, this depth was carried well away from the center and then reduced rather abruptly to match with the side-rail depth. Although this shape was not correct from the standpoint of uniform stress conditions, from the standpoint of rigidity it served two purposes: the average depth of the beam was increased and the flange curvature was confined to a small area so that any weakening effect would be local and not weaken the beams throughout their lengths.

The inner ends of the four arms of this experimental member were brought directly in line, and the junction was made by arc-welding along with the use of two small diamond-shaped plates, these plates acting as local reinforcements to make up for the web cutout required for propeller-shaft clearance, and also acting as dams to facilitate deposition of weld material. The resultant joint was one of unbroken continuity with no points of flexibility to cause beam deflection.

As a preliminary step, the X-member was removed from the standard frame and replaced by the I-beam-type X-member. No other changes were made in the frame. The torsional stiffness was increased as follows: from rear end of

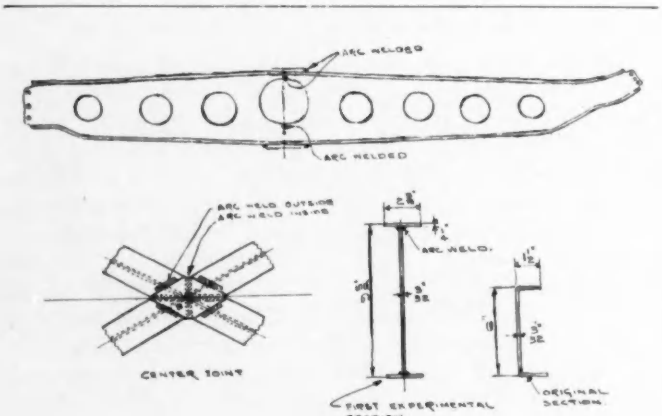

Fig. 4 – Experimental I-Section X-Member

X-member to front end of X-member, approximately ten times; from front end of X-member to front axle, approximately three times.

This frame was built into the car – a standard 1935 eight-cylinder model with independent front-wheel suspension, built-in trunk, and steel top – and various tests conducted. Many changes were made to the frame to vary its stiffness, and the effect of such changes on performance characteristics was determined. From this test work the specification and design of the final frame were arrived at. A comparison of the standard frame and the experimental frame follows. The two frames are shown in Fig. 5, the lower one being the standard frames.

Standard Frame

Side Rails	9/64-in. x 6-in. channels, 1 3/4-in. flanges.
X-Member	3/32-in. x 6-in. channels, 1 1/2-in. flanges.
Rear Kick-Up	Boxed in from rear end of X-member to gas tank, reinforcement 5/64 in. thick.
Front End of Side Rails	Boxed in from front end of X-member to front cross-member, reinforcement 3/32 in. thick.
Finished Weight	265 lb.

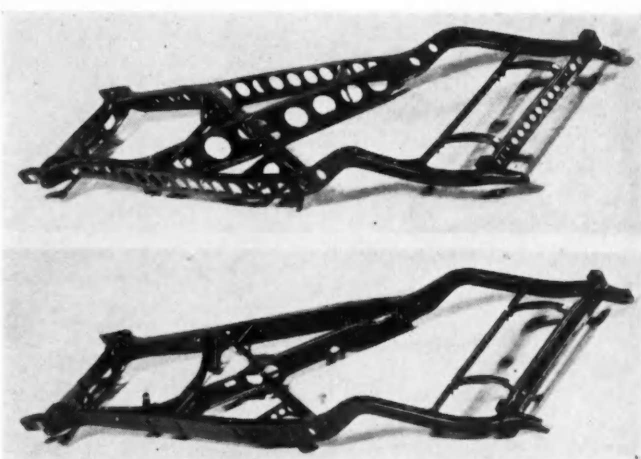

Fig. 5 – Standard Frame (*Below*) and Experimental Frame (*Above*)

Experimental Frame

Side Rails	3/32-in. x 7 3/16-in. channels (at dash), 1 3/4-in. flanges.
X-Member	3/32-in. web, 3/16-in. x 2 3/8-in. flanges, 10-in. deep at center.
Rear Kick-Up	No boxing, small reinforcement at center.
Front End of Side Rails	Boxed from front end of X-member to front cross-member, reinforcement 5/64 in. thick.
Finished Weight	246 lb.

The following figures indicate the actual torsional stiffness of the two frames. The figures are obtained by means of a test set-up as shown in Fig. 6, and are given in terms of ft-lb. torque to produce an angularity of 1 deg. between any two indicator beams.

	Rear of X-member to Front of X-member	Front of X-member to Front Axle Center Line
Standard Frame	1225 ft-lb./deg.	1360 ft-lb./deg.
Experimental Frame	7000 ft-lb./deg.	3400 ft-lb./deg.

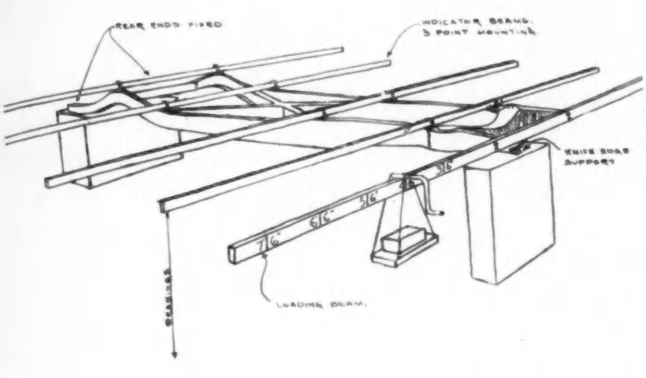

Fig. 6 – Torsion Test Set-Up

It will be noted that the experimental frame, instead of being heavier than the standard frame as originally expected, was actually 19 lb. lighter. Because of the great gains in stiffness accomplished with the I-beam X-member, it was decided to develop an I-section that could be produced cheaply enough for commercial purposes. Fig. 7 is an illustration of an actual test specimen as developed finally. The flanges are special rolled "T" sections, and the web is of ordinary flat stock blanked to shape. The flanges, after being formed to proper shape, are flash-welded to the web sections. Further test data must be prepared to determine closely the average gains in stiffness for a given weight of material by substitution of this section for the channel. However, in the standard frame previously specified, the X-member assembly weighed 51 lb.; in the experimental frame it weighed 78 lb.

In Fig. 3 is indicated the type of movement that takes place in the flange of the channel where the flange is curved. One of the reasons for the increased stiffness with the I-beam section is that this flange-buckling tendency seems to be greatly reduced. Referring again to Fig. 3, for the flange of the channel to deflect in a vertical plane, it is necessary that the beam shift sidewise, otherwise a shearing action would have to take place in the fibers adjacent to the web. Now, with the I-section, equal and opposite lateral resistance exists both sides of the web due to the section having double flanges. In consequence, the web maintains its alignment and, because of the resistance of the fibers to shear, the flanges remain in their proper location and perform their function of resisting tension and compression resulting from vertical bending loads. This factor, at least, is thought to be the cause of the reduced buckling action. However, further checking must be done.

Fig. 7 – I-Section Developed for Test Specimen

The Frame and Body Combination

When the frame is attached to the body structure in the manner generally in use, that is, with hard or semi-hard shims resulting in a rigid connection, the action of both units when subjected to torsional loads is modified by the other. In consequence, the torsional stiffness of the combination is not equivalent to the sum of the stiffness values of each unit individually.

In Fig. 8 is shown diagrammatically a cross-section of a frame and body. Either unit, twisted individually, distorts along the lines of least resistance, thereby determining the axis of twist, designated as passing through points *A* and *B*. When the two are tied together, the action of each is modified by the resistance of the other to this particular action. As a result, both structures are forced to distort in a manner to which they have greater resistance, and the resultant axis of the combination may be anywhere between points *A* and *B*, as for instance, point *C* shown in the illustration. In consequence, as the frame's stiffness is increased, axis *C* in the combination is brought closer to *B*; the frame's contribution to the overall stiffness of the assembly is more nearly the same as its individual stiffness and the body, because of the greater distance between *A* and *C*, contributes a greater degree of rigidity than it did with the weaker frame. In other words, if the frame's stiffness were to be stepped up, say 1000 ft-lb., the increase to the assembly would be in excess of that amount. The following figures indicate the degree to which this statement is so:

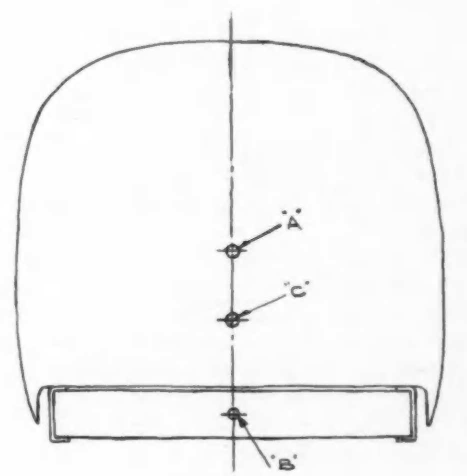

Fig. 8 – Cross-Section of Frame and Body Showing Axes of Twist

The stiffness of the standard car, from rear end of X-member to front end of X-member, was 10,330 ft-lb. per deg. The stiffness of the frame alone was 1225 ft-lb. per deg. By increasing the frame stiffness to 7720 ft-lb. per deg., the stiffness of the car was increased to 20,060 ft-lb. per deg. Although the frame increase amounted to only 6495 ft-lb. per deg., the increase in the assembly was 9730 ft-lb. per deg.

The Rear Kick-Up

Although the stiff frames experimented with created an increased feeling of stability throughout the entire car, the most noticeable change was a very marked reduction in tail-

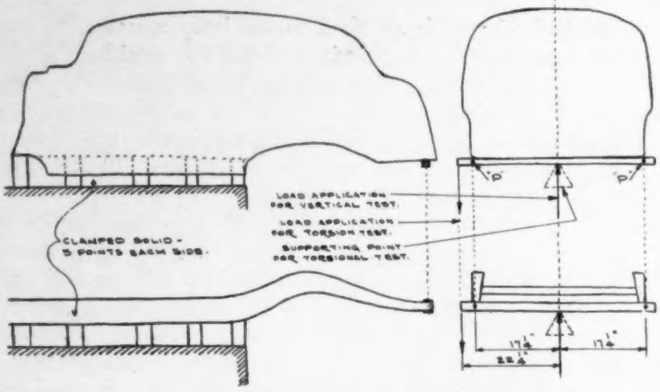

Fig. 9 – Frame and Body Test Set-Up (See Fig. 10 for Results)

end shake. Increases in the stiffness of the X-members seemed to have a direct and almost proportionate effect on the degree of movement occurring in this region when the car was driven over rough pavements. This observation was extremely interesting as earlier experiments had been conducted in an attempt to improve conditions at this point, but without much success. These experiments had been made on the rear kick-up of the frame to determine if increases in rigidity at this point would show an improvement. The result of this work had led to the belief that an increase in the vertical stiffness of the side rails through this section was of little benefit, but that boxing of the kick-up, causing it to be resistant to torsion, resulted in some improvement. The reason for this improvement had not been determined.

Upon observing the effect of the stiff X-members, it was decided that certain tests should be made to obtain information as to the relative stiffness of frame and body at the rear when stiffness was added at the front. Accordingly, a test set-up was made that exaggerated the condition arising when stiffer X-members were used. As illustrated in Fig. 9, each unit was mounted on a surface plate, the rear end overhanging from rear end of X-member back. Torsional and vertical loads were applied, with results as shown by the curves in Fig. 10. From a study of these results an analysis of the rear-end condition was set up about as follows:

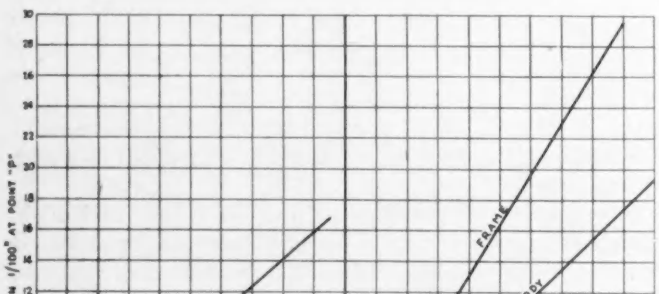

Fig. 10 – Stiffness Curves from Test Shown in Fig. 9

In a conventional car with comparatively weak X-members, torsional body deflection takes place somewhat as shown diagrammatically at *A* in Fig. 11, the deflection being greatest at the rear because of the higher frame stiffness at the forward end of the body. This higher stiffness is due to the boxing of the front end of the frame. It will be noted that, as shown, the body deflection takes place due to deflection in the body corners, the top, floor, and sides moving in relation to each other as units.

The boxing of the rear kick-ups can be likened to the addition of a small torsion member to the body as indicated by *B*. Naturally, this addition will influence the stiffness of the body, the magnitude of its effect being determined by its torsional value and the efficacy of its attachment. However, in the proportions shown, its effect must be small.

An increase in the kick-up beam's stiffness can have practically no effect in the conventional set-up as it would merely

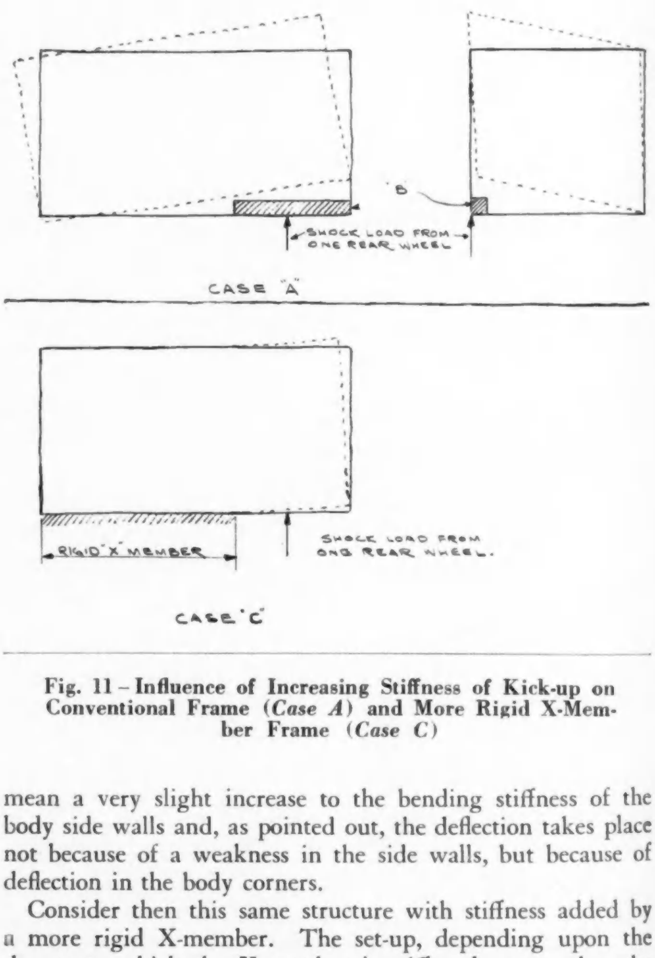

Fig. 11 – Influence of Increasing Stiffness of Kick-up on Conventional Frame (*Case A*) and More Rigid X-Member Frame (*Case C*)

mean a very slight increase to the bending stiffness of the body side walls and, as pointed out, the deflection takes place not because of a weakness in the side walls, but because of deflection in the body corners.

Consider then this same structure with stiffness added by a more rigid X-member. The set-up, depending upon the degree to which the X-member is stiffened, approaches the condition shown at *C*, Fig. 11; the corner deflection in the body is eliminated over the area encompassed by the four ends of the X-member, and deflection takes place only from the X-member back. Torsional loads applied at the rear are resisted by the vertical bending stiffness of the body side walls, these acting as extensions from the portion of the body supported by the X-member.

Under these conditions vertical stiffness added to the frame kick-up would cause a further increase in tail-end rigidity. However, as evidenced by the curves in Fig. 10, the body rigidity is already several times the rigidity of the frame, so

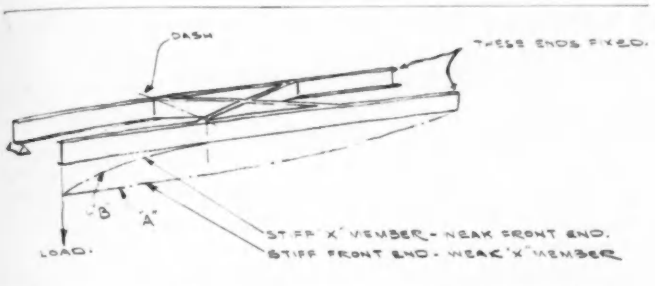

Fig. 12 – Deflection Characteristics – *A*, Stiff Front End, Weak X-Member – *B*, Stiff X-Member, Weak Front End

great increases would be necessary to obtain any large percentage of improvement. Furthermore, the curves are taken from an ideal structure with X-member deflection at zero, a condition that cannot be obtained in actual practice.

Front End of Frame

The stiffness of this portion of the frame is of importance as its vibrational frequency has a decided effect on the dynamic problems associated with the front-wheel suspension, flexibly mounted motor, and so on. Apparently the vertical stiffness of the side rails is of major importance, experiments indicating that the lateral stiffness does not have nearly as much effect in quieting the front end.

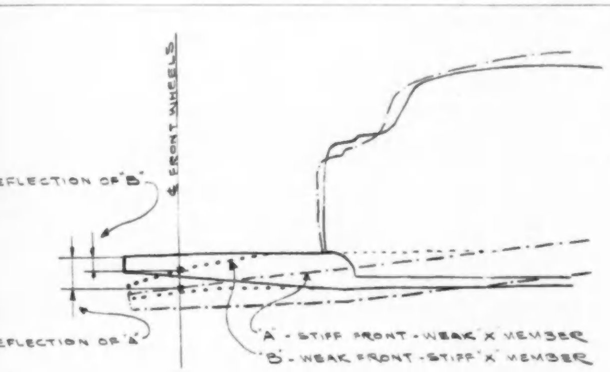

Fig. 13 – Deflection Characteristics – *A*, Stiff Front, Weak X-Member – *B*, Weak Front, Stiff X-Member

With the conventional frame having comparatively weak X-members and heavily boxed front side-rail extensions, the front end of the frame, in many cases, is stiffer torsionally than the portion to rear of the dash. In conditions of this kind the side-rail section to the rear acts somewhat as a cantilever supported by the forward end of the side rail, deflection taking place as indicated at *A* in Fig. 12. With a stiff X-member as experimented with, this condition is reversed, the front-end extensions functioning as cantilevers supported at the dash, deflection occurring as at *B*.

A check on the stiffness of the frame alone is useful only for making design comparisons. The actual overall result must be obtained by a check on the frame and body assembly. It is possible, for instance, to add stiffness to the X-members and have greater stiffness in the car from dash to front axle, even though the side rails are actually weaker through this section. In Fig. 13, *A* is a standard frame, its deflection characteristics when subjected to a torsional load being as shown. *B* is the same frame except that the X-member stiffness has been greatly increased and the boxing at the front end has been removed. It will be noticed that, although the actual deflection of the rail ahead of the dash is greater in the case of *B*, the stiffening of the section back of the dash, in effect, cuts down the effective length of the spring; the total deflection at the front axle, therefore, is less.

Body Attachments

A torsionally rigid frame is of little avail if the body attachment is not made properly. After all, the main function of the chassis is to support the body weight properly, and it is this weight primarily that is responsible for the torsional loads which are imposed upon the frame structure; a chassis without the body, driven over rough pavement, would be punished very little.

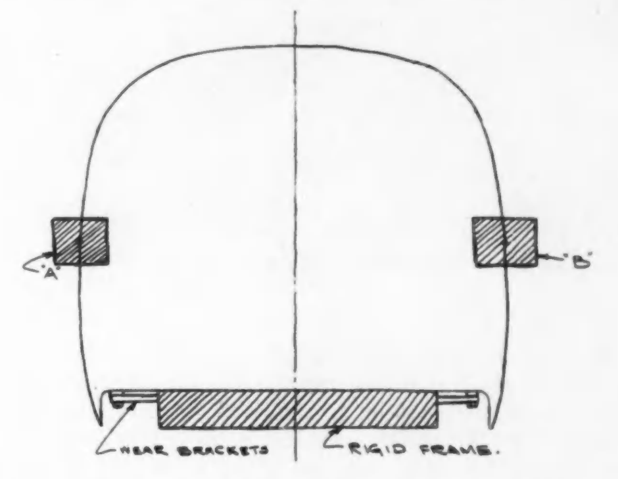

Fig. 14 – Infinitely Rigid Frame with Weak Supporting Brackets

In Fig. 14 is shown a cross-section of a frame which, it is assumed, is infinitely rigid. From the frame, brackets extend outwardly to act as supports for the body. The brackets shown are not capable of carrying the body weight, designated as being concentrated at *A* and *B*, without deflection, either because of weaknesses in the design of the brackets themselves or because of inadequate attachments to the frame. As a result the brackets react as springs and, in conjunction with the body mass, objectionable natural frequencies are set up. Considering, therefore, the four X-member extremities as being fixed points, it is necessary that the body be sup-

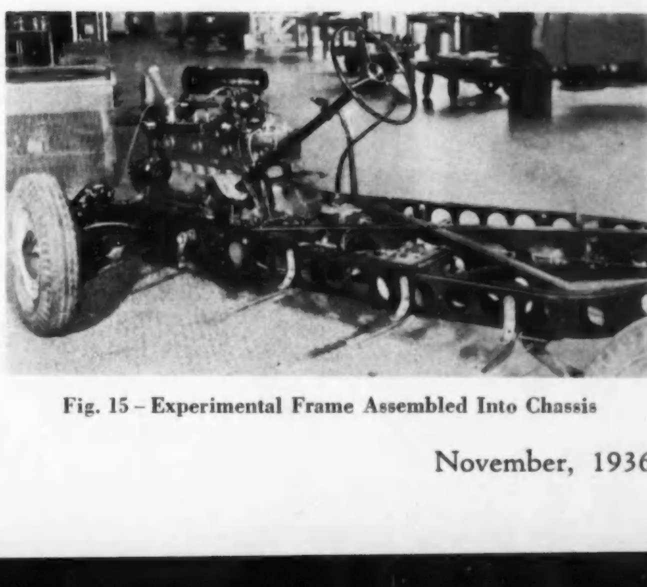

Fig. 15 – Experimental Frame Assembled Into Chassis

ported adjacent to these points and that the supports be of an efficient nature. Intervening supports, of course, are used as required for stopping body-sill deflection at the door openings.

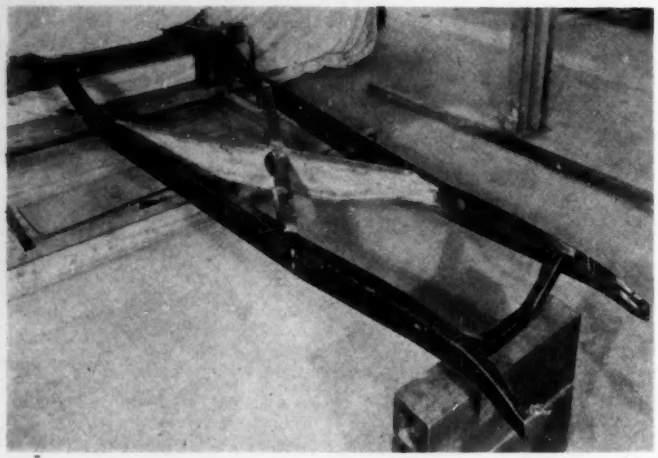

Fig. 17 – Experimental Frame with I-Section X-Members

Summary

Fig. 15 shows the experimental frame assembled into the chassis. Fig. 16 shows the relative stiffness of various standard frames as compared to replacement frames having an I-beam type X-member. Fig. 17 is an experimental set-up consisting of an X-member, side rails, and front cross-member. The X-member is 8 in. deep, with 1/8-in. thick flanges, 2 3/8-in. wide, and 3/32-in. web. The weight of this structure is 145 lb., its torsional stiffness is given in the chart, Fig. 16, No. 8.

This paper has dealt entirely with the frame as a torsionally resistant member. It is not contended, however, that the torsional stiffness of a frame is a direct measure of its excellence. As a matter of fact, much remains to be determined concerning the vibrational problems connected with the structure. All indications are, though, that the rigid frame is fundamentally desirable and that it has possibilities for improved riding qualities that we can never hope to reach with the weak frame.

Stressed-Skin Structures for Aircraft

(*Continued from page* 447)

flange must be internal rather than external as used on the wings.

A few years ago it was considered that maintenance and repair costs on an airplane employing such construction would be almost prohibitive. However, the experience obtained by Transcontinental & Western Air, Inc., in the operation of a fleet of airplanes having this type of structure indicates that the normal life which may be expected of it, barring accidents of any nature, is on the order of from 10,000 to 15,000 hr. flying time. Therefore, maintenanace costs due to the wear and tear of service are reduced to a minimum, and it has been found that, even when damaged by minor accidents, the structure can be repaired very easily.

One very interesting characteristic of a stressed-skin structure is its ability to carry load after actual failure in bending has occurred. The gage of fuselage and wing skin and the type of spacing of stiffeners are determined by the compression loads due to bending. Failures obtained during static load test of various structures invariably occur in the part of the structure which is under compression load and, after a permanent failure has been obtained, the structure will still carry approximately 75 per cent of the total load required to obtain the failure. Due to this characteristic there have been cases where, after a fuselage was buckled badly due to excessive tail-wheel loads obtained in bad landings, the airplane was flown to a repair station after hurriedly bumping out the worst of the buckles in the skin. The same thing has been done where wings have been damaged by fence posts in landing in short fields. This is a very desirable feature for the operator, either commercial or military as, in a great many cases, it obviates the necessity for making field repairs or disassembling and trucking the airplane back to a repair base.

In view of the ruggedness of this type of structure, its adaptability to quantity production, and the reduced maintenance cost for the operator, it is believed that it has a decided advantage over any type of structure now in use.

TORSIONAL FRAME STIFFNESS
FOOT-POUNDS TORQUE FOR ONE DEGREE OF TWIST

NO.	NAME	WEIGHT, lb.	REAR OF 'X' TO FRONT OF 'X'	FRONT OF 'X' TO FRONT AXLE ℄
1	STANDARD X-MEMBER TYPE FRAME	265	1225	1360
2	SAME AS #1, BUT HOLES IN FRONT SUB-CHANNELS CLOSED	271	1250	1440
3	SAME AS #2, BUT FRONT MEMBER BOXED	276	1275	1575
4	SAME AS #1, BUT CHANNEL X-MEMBER REPLACED WITH I-BEAM X-MEMBER	302	12,225	4050
5	REPLACEMENT FRAME FOR #1, I-BEAM TYPE X-MEMBER	246	7000	3400
6	STANDARD X-MEMBER TYPE FRAME	212.5	440	760
7	REPLACEMENT FRAME FOR #6, I-BEAM TYPE X-MEMBER	173	3730	1690
8	SIDE RAILS & FRONT MEMBER FROM #6, I-BEAM TYPE X-MEMBER	145	3530	2090
9	STANDARD FRAME, TUBULAR TYPE	170	600	828
10	REPLACEMENT FRAME FOR #9, I-BEAM TYPE X-MEMBER	160	2450	2425

Fig. 16 – Relative Stiffness of Standard Frames and Those with I-Beam X-Members

Photo-Electric Combustion Analysis

By R. A. Rose, G. C. Wilson, and R. R. Benedict
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SOLUTION of the problem of igniting and burning the fuel in the high-speed Diesel engine profoundly affects its development, according to the authors.

This paper describes the photo-electric set-up selected to indicate the behavior of the fuel in the combustion-chamber because of its high speed, its intensity, its zero time lag, and its freedom from inertia effects.

A magnetic-type oscillograph for recording the impulses, a cantilever-spring indicator for picking up the pressure impulse, and an amplifier between the photo-cell and the oscillograph, comprise the principal parts of this instrumentation, as applied to a single-cylinder test engine.

Results of tests with a three-beam vibrator-type oscillograph are given with oscillograms for different fuels, loads, and injection angles.

Other tests are described using a cathode-ray oscillograph and a high-speed camera.

This high-speed indicating system, recording three simultaneous impulses, furnishes the means for studying a variety of combustion problems in both Otto- and Diesel-cycle engines.

Twenty-four varied Diesel fuels were studied in both the present test engine and in a commercial high-speed unit in the summer of 1936.

STUDIES of combustion in a high-speed internal-combustion engine are complicated by the fact that the entire process must be completed during a period of time which is less than 0.01 sec. The process of combustion is even more complicated in the Diesel engine than in the gasoline engine because, in the former, the liquid fuel is injected into the cylinder only a few thousandths of a second before combustion must occur; and, during the brief interval in which the piston approaches and passes through the dead-center position, the fuel must be injected with a pressure sufficient to cause atomization and effect proper penetration. At the same time the fuel must be mixed with the air in the combustion space, and the minute particles must vaporize, ignite, and burn at a rate that will build up pressure quickly but not so suddenly as to cause shock and roughness. There are many ways of partially controlling this action, and one of the secrets of this control lies hidden in the fuel itself. The source of the crude and the method of refining have a decided effect upon the behavior of the fuel. Boerlage and Broeze have pointed out that two substances may have the same chemical formula and yet behave very differently in the combustion-chamber because of a difference in molecular structure.[1]

High-Speed Diesel

The future development of the high-speed Diesel engine depends upon the successful solution of the problem of igniting and burning the fuel. That this type of engine does have a definite field for service and that it should give such service while using a fuel of not too elaborate requirements, seem to require no supporting arguments. Once the behavior of the fuel in the combustion-chamber is clearly understood, the refiner can prepare the fuel without undue expense.

Ignition Lag

Some time ago Ricardo explained combustion in the Diesel engine by dividing it into three distinct phases[2] as follows:

(1) A delay period, during which fuel is admitted but either no ignition takes place or the ignition is restricted to some very localized nucleus.

(2) The mechanical spread of flame accompanied by a rapid rise in pressure.

(3) The remainder of the fuel burns under direct mechanical control as it enters from the injection nozzle.

Ignition lag has been defined as the period of time that elapses between the start of fuel injection and the instant pressure starts to rise because of combustion. This delay period is so important that it seems worthy of more detailed study. With high-speed recording apparatus it is possible to subdivide this first phase into two periods:

(a) The delay between the beginning of injection of fuel and the time it ignites and starts to burn.

(b) The delay between ignition and the beginning of the pressure rise.

Back in 1927, Dr. G. G. Brown showed that the knocking

[This paper was presented at the Semi-Annual Meeting of the Society, White Sulphur Springs, West Va., June 2, 1936.]

1 See "Ignition and Combustion of Diesel Fuels", by G. D. Boerlage and J. J. Broeze, presented at the Annual Meeting of the A.S.M.E., Dec. 2-6, 1935.

2 See "The High-Speed Internal-Combustion Engine", by Harry R. Ricardo, 1931, D. Van Nostrand Co., Inc., New York.

tendency of the fuel in a gasoline engine seems to vary directly with the rate of rise of pressure accompanying the progressive explosion, and indirectly with the spontaneous ignition temperature (S.I.T.) of the fuel.[3] Even though fuel knock in the Diesel engine is basically different from detonation in the gasoline engine, the same two factors of S.I.T. and rate of pressure rise have an important effect upon combustion in the Diesel. In the Diesel the trouble arises during the first stages of combustion whereas, in the gasoline engine the detonation is set up by the last-to-burn portion of a combustible mixture.

Ignition-Lag Studies

The indicating system developed at the University of Wisconsin is expected to furnish information supplementary to that being gained at other institutions. The accomplishments of A. M. Rothrock, Dana W. Lee and their associates working under the National Advisory Committee for Aeronautics (N.A.C.A.) are outstanding.[4] They have shown by actual photographs how fuel jets break up and burn. G. D. Boerlage and J. J. Broeze of Delft, Holland, have connected ignition lag with the cracking tendency or thermal stability of several fuels.[1] They further point out that, in some instances, a close inspection of the diagram indicates that the pressure decreases at first and may increase appreciably before rapid combustion sets in. This point also has been mentioned by other experimenters. It seems that an extremely high-speed indicating apparatus might show these pressure changes more clearly. T. B. Rendel and his Volunteer Group for Compression-Ignition Research are making splendid progress in the matter of establishing a satisfactory procedure for rating Diesel fuels.[5]

The apparatus that this paper will describe is purely a research set-up which requires painstaking care to insure proper operation of all parts at the same time. When operating, many photographic records can be obtained from an actual engine operating at speeds up to 1200 r.p.m., or faster. Three impulses are obtained for a single cycle. At the present time, these impulses are obtained for:

(1) Movement of the needle-valve stem in the fuel-injection nozzle (valve stem).

(2) Radiation emission in the combustion-chamber.

(3) Pressure in the combustion-chamber.

The results obtained thus far are by no means conclusive. They are offered at this time as a progress report in order that criticism may aid in making improvements.

[3] See University of Michigan Research Bulletin No. 7, May, 1927; "The Relation of Motor-Fuel Characteristics to Engine Performance", by G. G. Brown.

[4] See N.A.C.A. Report No. 520, 1935; "A Comparison of Fuel Sprays from Several Types of Injection Nozzles", by Dana W. Lee.

[5] See S.A.E. TRANSACTIONS, June, 1936, pp. 225-233; "Report of the Volunteer Group for Compression-Ignition Fuel Research", by T. B. Rendel.

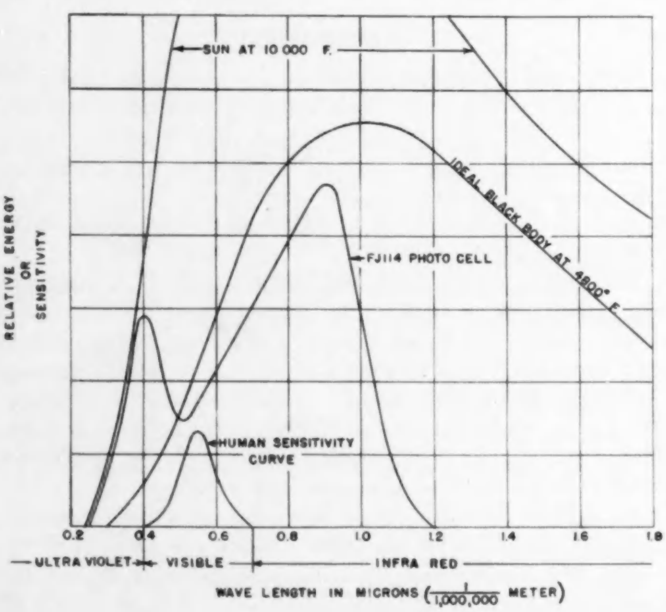

Fig. 1 – Sensitivity of the Photo-Cell

The Photo-Electric Cell

In searching for a high-speed means of picking up impulses, the vacuum-type photo-electric tube was selected as ideal. This tube is commonly referred to as a photo-cell. It has a high degree of sensitivity with a time lag of zero, and is entirely free from inertia effects. Its response is directly proportional to the light received, and it is perfectly stable in its operation and free from all hunting action.

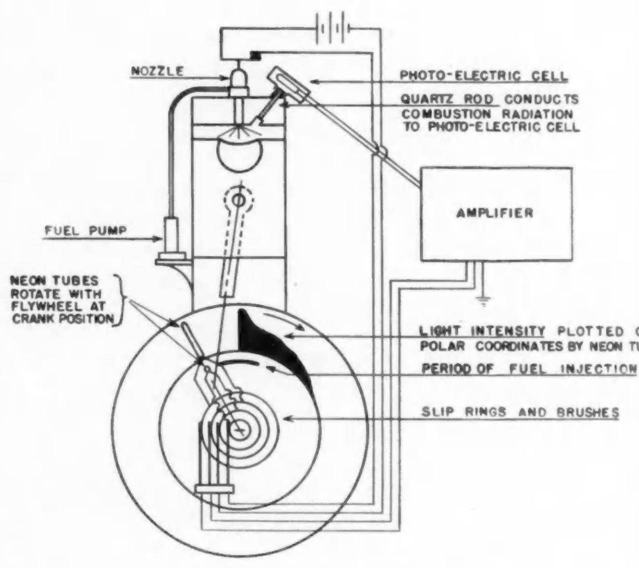

Fig. 2 – Neon-Tube Optical Indicator

The active surface of the photo-cells used for this indicating system is caesium, deposited on oxidized silver and enhanced by other special activation treatment. Because of its high sensitivity in the infra red, this photo-cell is especially useful in "invisible-light" applications. It is also a most efficient means for detecting a low-temperature light source. Fig. 1 shows, by curves, the sensitivity of this photo-cell as compared with human vision. The human eye can detect wave lengths ranging only from 0.4 to 0.7 microns, whereas the photo-cell is sensitive to wave lengths of 0.3 to 1.2 microns. In other words, the photo-cell is sensitive to both ultra-violet and infra-red radiations that are not visible. The other two curves, one for the sun and one for an ideal black body, show relative radiation energy and are included merely for purposes of comparison.

Neon Tubes

The first test results were obtained by allowing the photo-cell to look into the combustion-chamber through a quartz

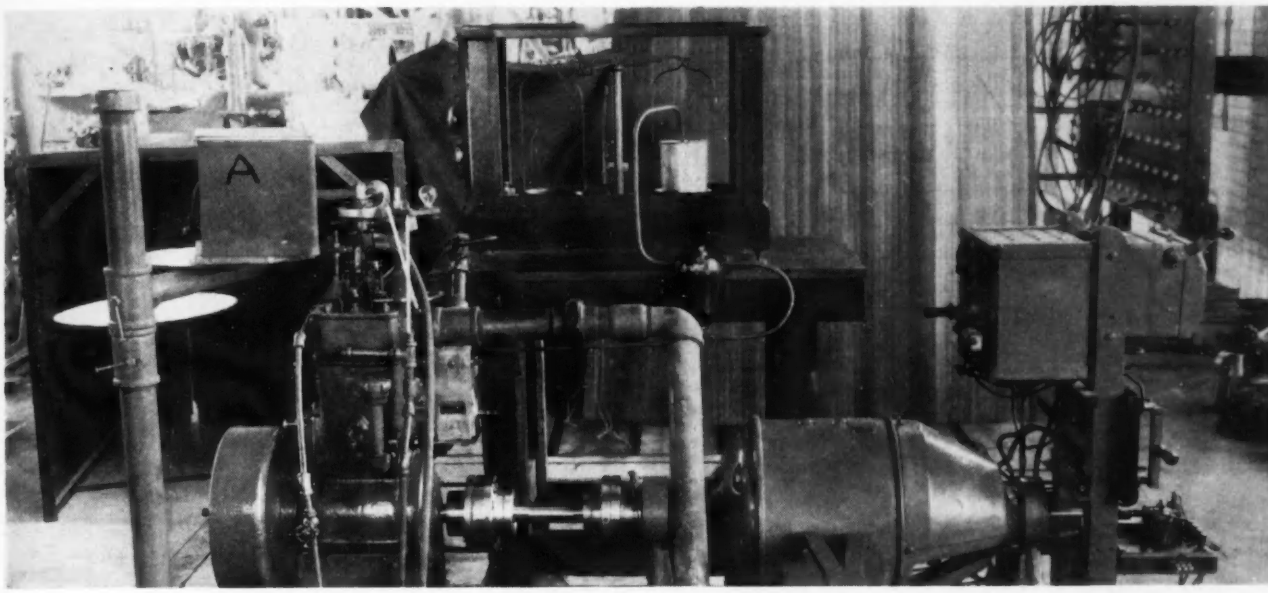

Fig. 3 – Test Engine and Dynamometer

rod that served as a window, as shown in Fig. 2. The current from the cell was amplified and delivered to a 3-in. neon tube (tune-a-light) which was mounted on the flywheel at crank position. This neon tube has the property of glowing along its length a distance proportional to the current received from the amplifier. Another small neon tube showed the period of time the valve stem was off its seat. When the pressure element was used with this set-up, another 3-in. neon tube was placed 180 deg. out of phase with the crank. This scheme furnished excellent optical diagrams on polar coordinates, but attempts to photograph these diagrams at a speed of 1200 r.p.m. were not successful. Both the photographic material and the vapor or gas in the tubes were varied in an effort to produce a light to which the film or plate was sufficiently sensitive. Although these possibilities were not exhausted, it was decided to use an oscillograph for recording the impulses in order to proceed with the major project.

The Pressure Element

A Maihak No. 4 cantilever-spring indicator was reconstructed to furnish the pick-up for the pressure impulse. A 1000-lb.-scale spring was used with a normal-size piston. The light-weight hollow piston together with the stiff spring has a very high natural period which overcomes the effect of inertia. The entire pencil motion was stripped back to the piston-rod to reduce the weight of moving parts to a minimum. The eye at the top of the piston-rod was made into a small light shutter. A uniform light focused on one side of this shutter is allowed to shine through to a photo-cell on the opposite side as shown in Fig. 3. In this way the photo-cell delivers a current proportional to the pressure in the combustion-chamber.

When the pressure-indicating system first was checked for accuracy, it was found that the high point of compression lagged behind dead-center position of the piston. The photocell and amplifier were checked independently from the pressure element and found accurate. After enlarging the connection to the indicator, the high point of the compression curve came exactly at dead-center.

Photo-Electric Cells with an Oscillograph

As stated previously, the neon-tube indicator was suitable for visual work but, for recording purposes, it was necessary to develop the system so that records could be obtained on a magnetic-type oscillograph. It was deemed desirable to retain the linear-response feature of the previous arrangement. Therefore, the amplifier between the photo-cell and the oscillograph was necessarily one that would cause little or no distortion of the record. Furthermore, in order to have a stable zero line, it was necessary to avoid the conventional type of amplifiers in which the low frequencies and "d-c" changes are not transmitted. The amplifier finally adopted was a direct-coupled or so-called "d-c" amplifier.

The general features of the amplifier are shown in Fig. 4. The first stage of amplification consists of a pentode amplifier. The pentode type of tube was chosen on account of its low-input capacitance, and the large-voltage amplification obtainable. Its heater is operated by alternating current from the same transformer which supplies the plate-supply rectifier circuit. The photo-cell supply voltage is obtained from the potential divider of the amplifier power supply circuit. Special attention was given to the shielding (see dotted lines in Fig. 4) and to the filter in the power-supply to eliminate interference. An unusual feature of the circuit of the first stage is the location of the ground. This feature will be discussed as follows:

Assuming the switch S in Fig. 4 to be thrown to the right, it will be seen that the first and second stages of the amplifier are coupled by the resistor R_2. That is, the plate current I_p of the Type "57" tube flows through the resistor R_2, thereby producing a voltage drop between the grid and cathode of the Type 2A3 tube. This voltage drop across R_2 is of the polarity shown in Fig. 4. Therefore, an increase in I_p causes the grid of the Type 2A3 tube to become more negative, and decreases the output current I_o. Summarizing the operation of the whole amplifier, it may be seen that an increase in the light to the photo-cell increases the current I_p, thereby making the grid of the Type 2A3 tube more negative, and thus decreases the output current I_o.

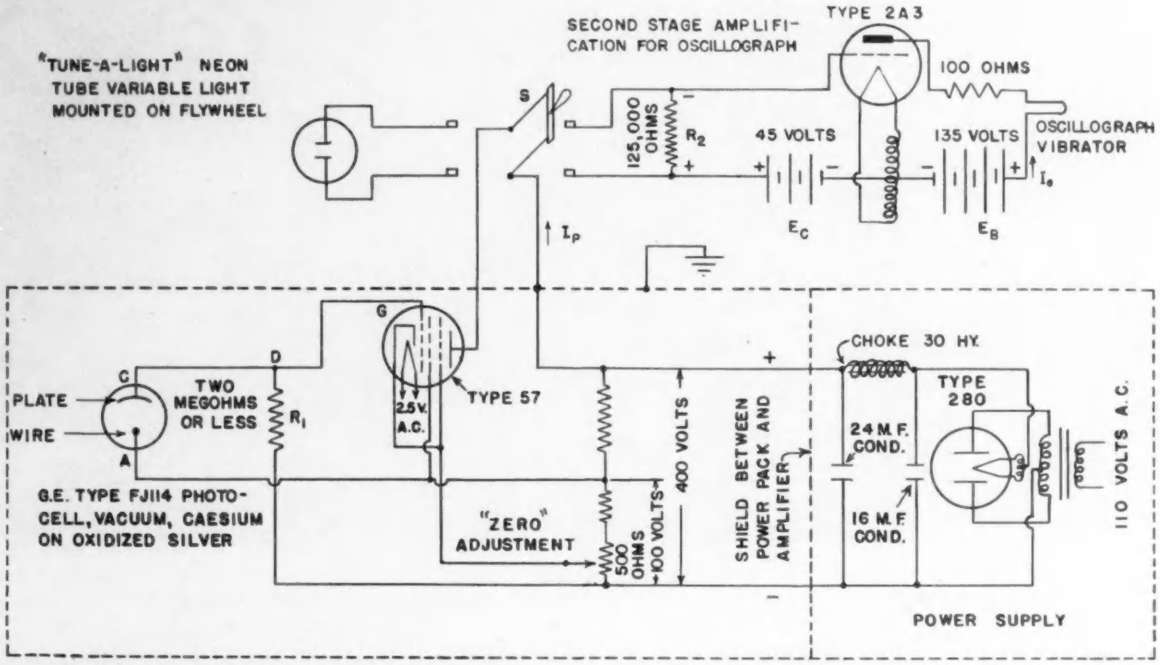

Fig. 4 – Photo-Electric Cell and Amplifier Circuit

The 45-volt battery E_c is connected with a polarity opposite to that ordinarily used for grid biasing. This arrangement is necessary to counteract partially the drop across R_2 occasioned by the normal value of I_p. This normal drop is 50 volts, so that the net normal grid bias of the Type 2A3 tube is about -5 volts. The corresponding output current I_o is approximately 0.120 amp. The deflections of the oscillograph vibrator are caused by decreasing values of I_o from a normal value of 0.120 amp. This scheme provides a practically linear response for deflections from 0.120 to 0.030 amp. (0.75 in. on the oscillograph), but has the disadvantage that the tube is overloaded. It is necessary to use care to see that the first stage of the amplifier is operating before switching on the second stage since the drop across R_2 is used as the bias voltage for the second tube.

The circuit is grounded at the positive terminal of the plate-supply circuit. This procedure eliminates 60-cycle interference which otherwise would originate in the filament-heating transformer of the Type 2A3 tube. This procedure also permits the use of common sources of voltage for the second stages when more than one amplifier is used. For example, in the tests to be described later, two independent first-amplifying stages were used on the pressure and radiation systems; but the two second-stage tubes of the amplifiers were supplied from the same filament transformer and used the same batteries E_c and E_b. In order to obtain steady voltage, it was necessary to use a storage battery for E_b.

The 500-ohm potential divider marked "zero adjustment" in Fig. 4 was used to adjust the normal deflection of the oscillograph vibrator. In the case of the pressure-recording system, this adjustment also permitted compensation for the initial light flux on the photo-cell, that is, the light flux with zero deflection of the indicator.

The accurate measurement of the ignition lag necessitates the reduction of time lags in the measuring system to a minimum. With this object in view, the grid resistance R_1 was limited to two megohms and the lead wire *CDG* was made short and of fine wire to reduce its capacitance to the shield and other wires. Similarly, the lead from the plate of the Type 57 tube to the grid of the Type 2A3 tube was one of low capacitance. Estimates of the speed of the amplifier response showed that the rapidity of response of the oscillograph vibrator was the limiting factor in the entire system. The standard vibrator of the Westinghouse three-beam oscillograph that was used, has a natural frequency of 5000 cycles per sec. (c.p.s.) without damping, and a sensitivity of about 120 mil. amp. per in. of deflection. The damping fluid used with the vibrators was such as to give a damping slightly less than critical. This damping was determined by a test of the deflection of the oscillograph vibrator as a function of the impressed frequency for an alternating current of constant amplitude. The deflection was constant up to a frequency of 1000 cycles per sec., which was about 15 per cent higher than the low-frequency value, and then started to decrease, dropping to 50 per cent of the low-frequency value at 4100 cycles per sec. and to 25 per cent at 5700 cycles per sec. The amplifier and oscillograph would, therefore, record a sustained wave up to an 800-cycle-per-sec. frequency without serious error in amplitude or phase. The response of the system to a sudden change would, however, be rapid enough to permit determination of the time of the change with an accuracy of 0.0001 sec. Figs. 6 to 9 show the type of records which were obtained.

To center the diagrams on the film, the film-holder shaft was run at half engine speed, and its speed and phase position were adjusted by means of a neon-tube stroboscope.

Test Engine

The engine used for these combustion studies is shown in Fig. 3, connected to an electric dynamometer. The box con-

Table 1 – Fuel Data

Fuel	Special Diesel	Gas Oil
Gravity, deg. A.P.I.	39.7	41.0
Flash Point, deg. fahr.	270	140
Fire Point, deg. fahr.	290	175
Viscosity at 100 deg. fahr., Saybolt Universal, sec.	43.5	33.0
Aniline Point, deg. fahr.	192	...
Sulphur, per cent	0.05	...
Distillation – Initial Point, deg. fahr.	500	355
5 Per Cent Point, deg. fahr.	526	...
10 " " " " "	536	400
20 " " " " "	543	418
30 " " " " "	548	428
40 " " " " "	556	438
50 " " " " "	561	449
60 " " " " "	568	460
70 " " " " "	578	475
80 " " " " "	590	496
90 " " " " "	612	510
95 " " " " "	630	...
Final Point, deg. fahr.	660	515
Viscosity-Gravity Constant (Moore & Kaye)	0.803	0.830
Cetene No. (Moore & Kaye)	70	62
Cetene No. (Waukesha)	70	60
Cetane No., C.C.R. method	73	60
Cetane No., delay method	62	50

taining the photo-cell and amplifier for the pressure impulse is shown in position at *A*.

A cross-section of the engine is shown in Fig. 5. The engine was constructed by redesigning certain parts of a gasoline engine, as follows:

A liner was placed in the cylinder to reduce the bore from 3¾ in. to 3⅜ in. The stroke of 4½ in. was left unchanged. Although originally the engine delivered 4½ hp., it was decided to consider 3½ hp. as full load in order to simplify matters for test purposes. A new four-valve head was designed to replace the original L-head. The arrangement of the four valves in the head permitted locating the injection nozzle at the center. A Bosch system of fuel injection is in use. Several types of nozzles are on hand, but the work with the vibrator-type oscillograph was limited to a pintle-type nozzle with an included angle of 8 deg. Recently, some oscillograms have been obtained while using a four-hole (each 0.005-in. diameter) nozzle. These holes are drilled such that the included angle is 30 deg.

The compression pressure of this engine is 565 lb. per sq. in., as determined by a calibration of the indicator-spring deflection. The valve-stem lifting pressure was set at 3150 lb. per sq. in. Before the connection was made for the pressure indicator, the entire combustion space was concentrated in the center of the piston. This center portion is of cast iron for retaining the heat, whereas the remainder of the piston is aluminum to reduce weight. Connecting the pressure indicator introduced additional clearance space in the form of an air cell, which made it necessary to fasten a stainless-steel plate to the top of the piston in order to restore the desired compression pressure.

Fuels

This first series of tests have been limited to two fuels. One is a special Diesel fuel of cetane number 73, whereas the second fuel is a gas oil (marketed as furnace oil) which has a cetane number of 60 as rated by the "C.C.R. method." The "delay method" gives values of 62 and 50 cetane numbers, respectively. Complete data for these two fuels are given in Table 1.

Results of Test with Three-Beam Oscillograph

Fig. 6 shows two oscillograms obtained while using the same fuel with injection timed to within 1 deg. of each other, the upper diagram for "no load" and the lower one for "full load". The three impulses are as follows:

Top. – A 60-cycle sine wave interrupted at bottom dead-centers for the 360-deg. interval, and at the start of movement of the valve stem to indicate the beginning of injection. This sine wave also furnished a means for measuring time interval. Top dead-center position was determined by bisecting the 360-deg. interval.

Center. – The pressure variation. Where the hump of compression is apparent, the fact that it coincides with the dead-center mark seems to prove the accuracy of the pressure-indicating mechanism.

Bottom. – Radiation.

Fig. 7 shows two full-load diagrams for Diesel fuel, the top one with injection 7 deg. earlier in the cycle than in the lower one. Fig. 8 compares gas oil with Diesel fuel while the engine was operating at full load of 3½ hp., but with a difference of 2.3 deg. in fuel timing. Fig. 9 shows two no-

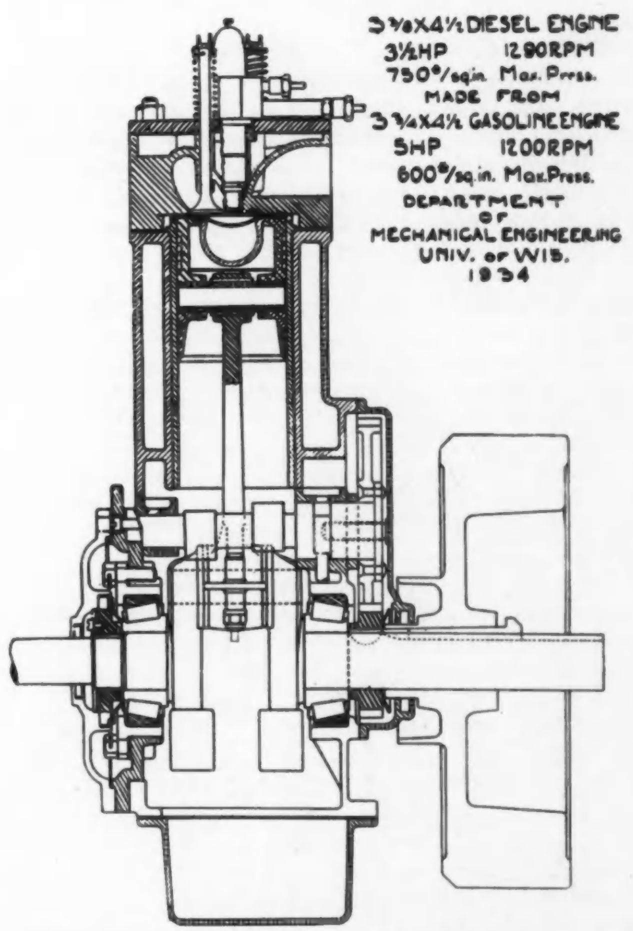

Fig. 5 – Cross-Section of Test Engine

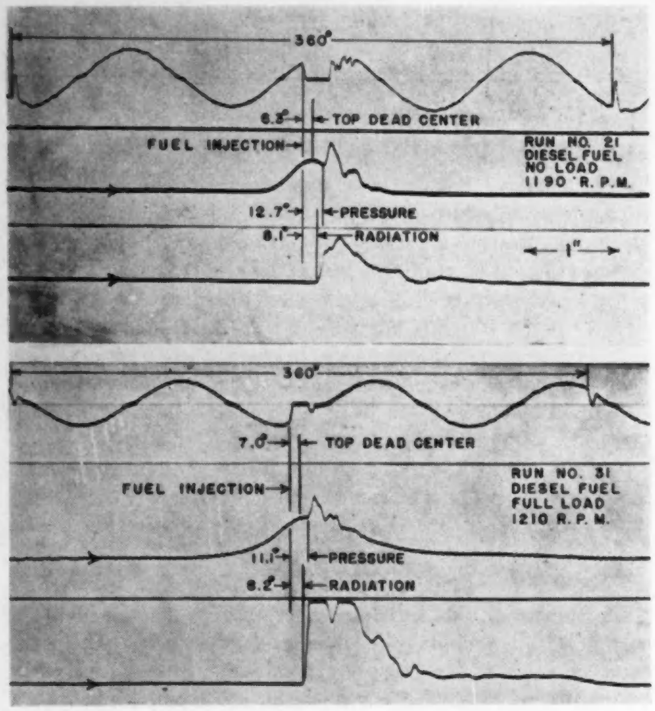

Fig. 6 – Oscillograms for Diesel Fuel, No Load (*Above*) and Full Load (*Below*)

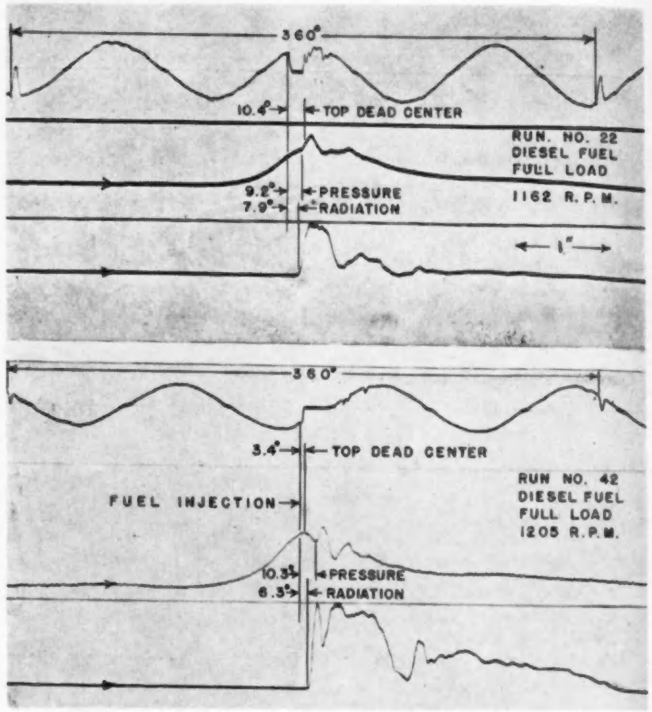

Fig. 7 – Oscillograms for Diesel Fuel, Full Load

(The top one with injection 7 deg. earlier than the bottom oscillogram)

load diagrams for gas oil while using radiation filters. In the upper one, the fuel was injected 4.1 deg. before top dead-center and, in the bottom one, it was injected 4.1 deg. after dead-center. For the upper diagram a visible filter was placed between the quartz window and the photo-cell, whereas for the lower diagram an infra-red filter was used. This diagram shows that both visible and infra-red radiations

Table 2 – Ignition-Lag Data from Vibrator-Type Oscillograms

		TEST NO.	INJECTION ADVANCE ANGLE	LAG AFTER INJECTION IN 1/1000 SEC. RADIATION	PRESSURE	DIFF.	FILTER
GAS OIL	NO LOAD	34	−6.5	1.5	2.0	0.5	INFRA RED
		36	−6.1	1.6	2.3	0.7	VISIBLE
		35	−8.1	1.8	2.3	0.5	INFRA RED
		37	−4.1	1.2	1.8	0.6	VISIBLE
		10	−2.6	1.3	1.3	0.0	
		38	+1.5	1.1	1.6	0.5	
		50	+4.1	0.8	2.0	1.2	INFRA RED
		51	+5.7	0.6	1.1	0.5	VISIBLE
		49	+8.7	0.5	1.2	0.7	ULTRA VIOLET RED PURPLE
	FULL LOAD	12	−4.6	1.2	1.6	0.4	
		13	0.0	1.1	1.5	0.4	INFRA RED
		40	+1.2	1.1	1.5	0.4	
		11	2.3	0.8	1.7	0.9	
		39	8.0	0.3	1.4	1.1	
DIESEL FUEL	NO LOAD	30	−9.5	1.5	2.1	0.6	
		21	−6.3	1.1	1.8	0.7	
		44	−6.1	1.2	2.0	0.8	
	FULL LOAD	23	−20.0	1.7	2.1	0.4	
		32	−11.6	0.9	1.5	0.6	
		22	−10.4	0.9	1.3	0.4	
		25	−9.5	1.1	–	–	
		31	−5.4	1.1	1.5	0.4	
		42	−3.4	0.9	1.4	0.5	
		41	−1.1	0.6	1.2	0.6	
		20	+1.1	0.5	0.9	0.4	
		14	+2.3	0.3	0.9	0.6	
		17	+2.8	0.3	1.1	0.6	
		18	+2.9	0.5	1.0	0.5	RED PURPLE
		15	+4.7	0.4	0.8	0.4	INFRA RED

were present in combustion. On another test for which the oscillogram is not included, a small amount of ultra-violet radiation was indicated.

Notice that the radiation always starts abruptly and, hence, its point of beginning is much easier to determine than that of the beginning of pressure rise caused by combustion. Notice also that the beginning of radiation is always definitely ahead of the pressure. In run No. 50, Fig. 9 (below), a faltering action is apparent in the radiation impulse. This action may be accounted for by the behavior of the valve stem as indicated by the slight interruption in its impulse line, or it may be due to a blowing-out action of the flame. The result is an unusually long delay in the pressure rise, which delay is indicated by a large difference in time between start of radiation and beginning of pressure rise as shown in Table 2. With the exception of a few such instances, this difference seems to be fairly constant and approximately 0.0005 sec.

These data, and the curves shown in Figs. 10 and 11, are not offered as conclusive evidence. However, the points do show a definite trend. The scale of the original oscillograms is such that 1 in. on the film represents approximately 0.0083 sec. In the case of the Diesel fuel, Fig. 10, the no-load points seem to fall on separate curves from the full-load data. This condition is not apparent in the case of the gas oil, Fig. 11.

Operating conditions varied somewhat during these tests. The jacket-water outlet temperature varied between 64 and 83 deg. fahr. for the gas-oil tests, and between 62 and 76 deg. fahr. for the Diesel-fuel tests. Exhaust-gas temperatures ranged from 225 to 445 deg. fahr. on no load, and 645 to 800 deg. fahr. on full load. All tests were made at speeds between 1150 and 1300 r.p.m.

The manner in which the trend of the points for both Diesel fuel and gas oil seem to indicate still shorter ignition lag for fuel injection later than top dead-center is not in agree-

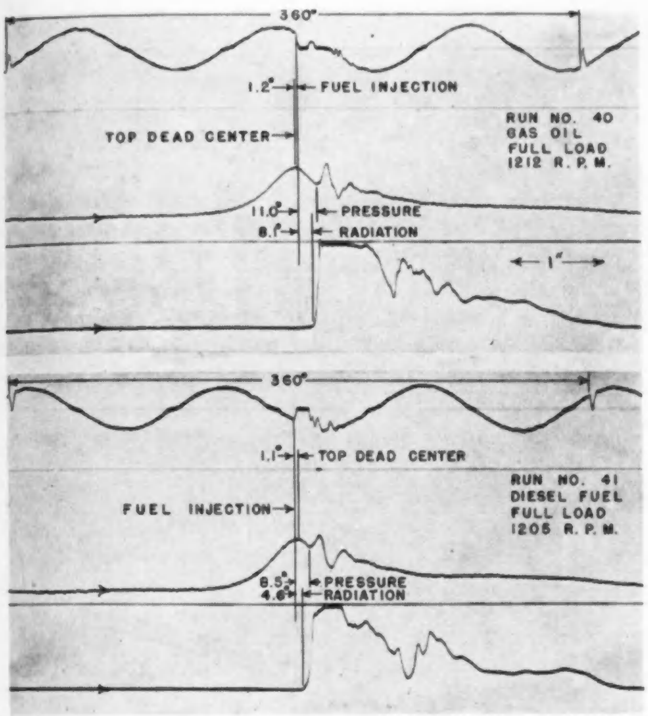

Fig. 8 – Oscillograms for Gas Oil (*Above*) and Diesel Fuel (*Below*), Full Load

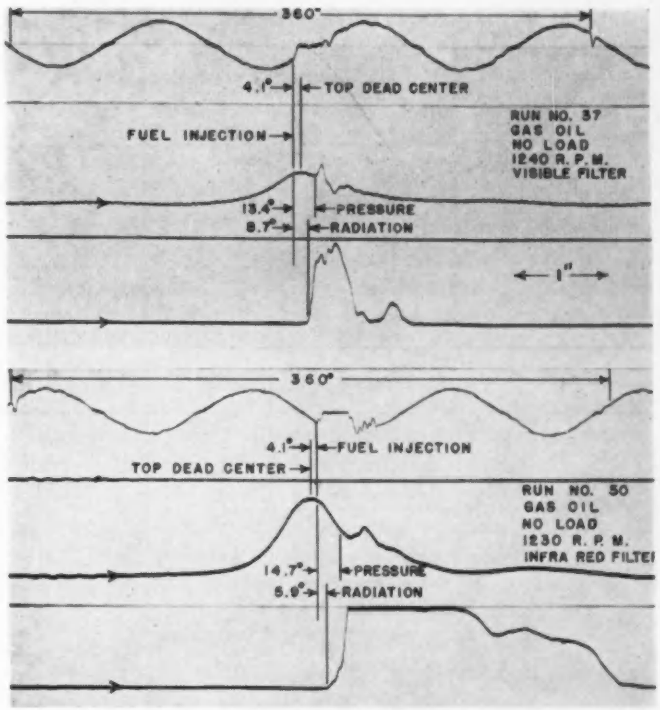

Fig. 9 – Oscillograms for Gas Oil, No Load, Using Radiation Filters

(*Above*) Injection 4.1 deg. before top dead-center with visible filter.
(*Below*) Injection 4.1 deg. after top dead-center with infra-red filter.

ment with results obtained by other experimenters[6] and is contrary to reason. This result seemed to point to inaccuracies in the method of indicating the beginning of fuel injection; therefore, a more accurate and complete indicating system for the fuel injection was provided.

The Present Indicating System

A diagrammatic sketch of the present set-up is shown in Fig. 12. For convenience of handling films, the camera, oscillograph, and developing trays have all been housed in a dark room. The radiation and pressure pick-up at the engine are the same as previously described. At the top of the nozzle valve stem is mounted a small shutter which has a light source on one side and a photo-cell on the other. Thus the entire movement of the valve stem is indicated. Any tendency of this valve to re-open for any reason, such as fuel-line surges, can be detected by this arrangement. The impulses from the three photo-cells are carried through amplifiers to a three-tube cathode-ray oscillograph.

Cathode-Ray Oscillograph

The cathode-ray oscillograph consists of three RCA oscillograph tubes, Type 907, with suitable power-supply equipment. These tubes have a fluorescent screen 5 in. in diameter upon which an electron beam impinges, giving a brilliant spot of light. This spot can be deflected by means of a voltage applied to a pair of deflecting plates within the tube, giving a useful deflection of about 4 in. The fluorescent screen material has a very short persistence time and produces a highly actinic spot which permits taking records with the high-speed moving-film camera described later. The oscillograph tubes all receive filament and anode power from the same supply circuit. The circuit for an oscillograph-tube connection has been shown in various publications.[7] The anodes are operated at 2000 volts, and the focusing electrode and grid potentials are adjusted to give a suitable fluorescent spot.

The photo-cell amplifier shown in Fig. 4 was modified so as to deliver the desired voltage to the deflecting plates of the cathode-ray tubes. The deflection of the spot is directly proportional to the deflecting voltage and, if there is no distortion in the amplifier, it is also directly proportional to the light on the photo-cell. Referring to Fig. 4, the second amplifier stage was removed and a 400,000-ohm resistor was connected in the plate circuit of the Type 57 tube. The voltage drop across this 400,000-ohm resistor was impressed on the cathode-ray tube deflecting plates. A biasing voltage was included in the connection to the deflecting plates, not only to compensate for the normal initial voltage output of the amplifier but also to give the spot on the cathode-ray tube an extra initial deflection in the negative direction. In this way the initial position of the spot was at one side of the tube, and about 3 in. of the tube surface was available for useful deflections. The deflections of the three tubes shown in Fig. 12 were perpendicular to the direction of motion of the film in the camera. The type of record obtained is shown in Figs. 13 and 14.

Since the cathode-ray oscillograph has no inertia, the major source of time lag in the previous equipment was removed, and it became necessary to re-examine the speed of response of the indicating system. The speed of response of the amplifier was increased by reducing the grid resistor from the Type 57 tubes to reduce their input capacitance. The capacitance

[6] See N.A.C.A. Report No. 525, 1935; "Some Effects of Injection Advance Angle, Engine-Jacket Temperature, and Speed on Combustion in an Internal-Combustion Engine", by A. M. Rothrock and C. D. Waldron.

[7] See *Automotive Industries*, March 14, 1936, pp. 398-402; "Engines on the Spot", by P. M. Heldt; also see Bulletin, Technical Series TS-2, R.C.A. Mfg. Co.

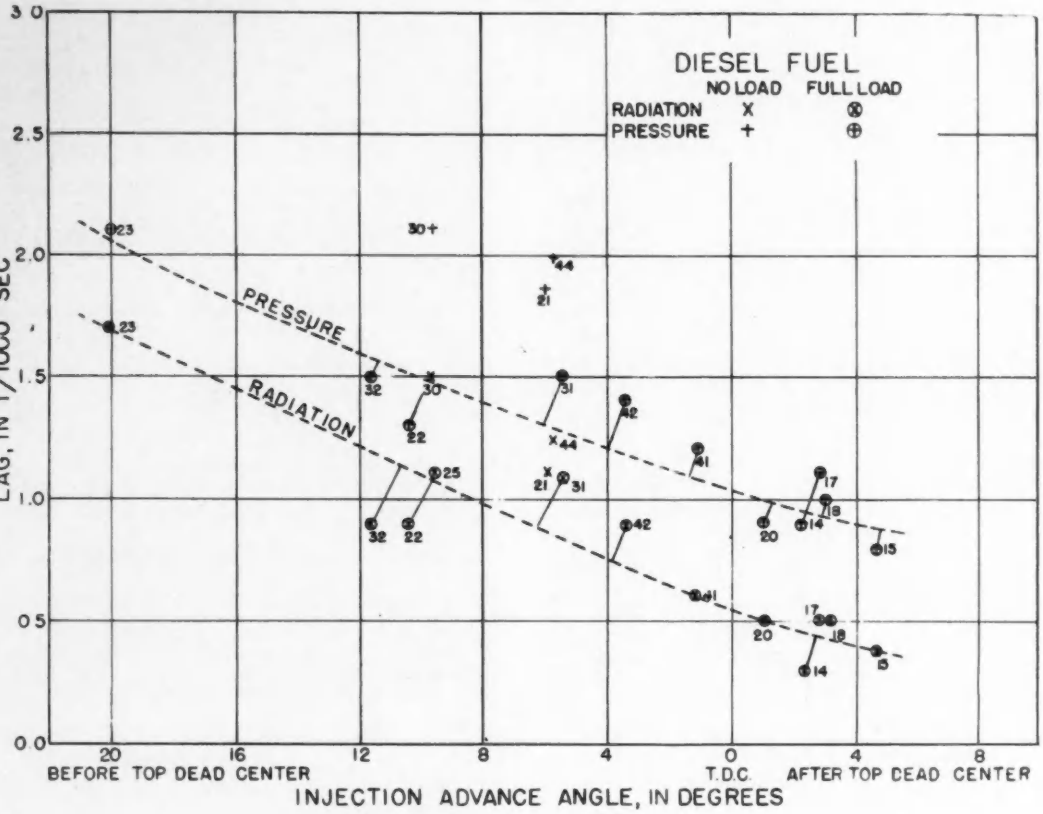

Fig. 10 – Ignition-Lag Curves for Diesel Fuel

of the lead wires from the amplifiers to the cathode-ray tubes was kept small. A test of the rapidity of response of the photo-cell amplifier and oscillograph was carried out as follows:

A disc containing three narrow radial slots was rotated

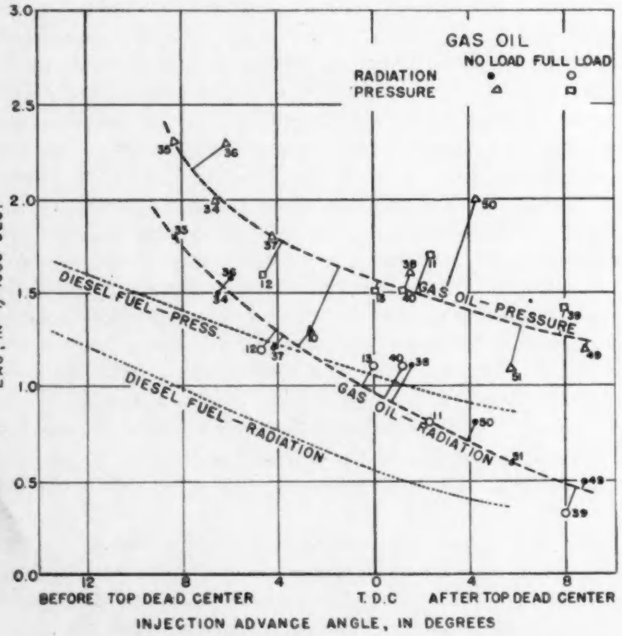

Fig. 11 – Ignition-Lag Curves for Gas Oil as Compared with Diesel Fuel

between a light source and the photo-cell in such a manner as to allow pulses of light to fall on the photo-cell. By running the disc fast or slow, the duration of the pulses could be changed, but their wave shape remained the same. Slow pulses lasting about 0.005 sec. were compared with fast pulses lasting 0.00035 sec. The shape of the pulse as seen on the cathode-ray oscillograph remained the same for the two speeds. Furthermore, the peak amplitude of the fast pulses was only 9 per cent less than that of the slow ones. These tests indicate that the amplifier and oscillograph will record the times of sudden changes in light to an accuracy of 0.00003 sec.

The cathode-ray tubes are operated at a low spot intensity ordinarily, but the spot intensity must be increased during the short time that the record is being taken. At present this control is obtained by manually adjusting a potential divider controlling the grid voltages of the three tubes. An attempt to use relays to perform this function was abandoned on account of disturbances which the relays caused in the position of the light spot.

High-Speed Camera

The camera consists of an aluminum drum, 11.4+ in. in diameter and 20 in. long, adjustable on its shaft and mounted in a light-tight box. Sheet film (20 × 24 in.), mounted on the drum, permits taking a dozen sets of graphs with one loading. Supersensitive plenachrome film has been found to be the fastest combination with the blue cathode-ray fluorescent spot. The lens is a f/1.5 – 3-in. moving-picture lens. The diameter of the drum was made such that the circumference at the surface of the film is 36 in. This fur-

nishes a convenient scale of 1 in. equals 10 deg. of crank travel which, for an engine running at 1200 r.p.m., corresponds to approximately 0.0014 sec. This arrangement is six times as fast as the previous set-up using a vibrator-type oscillograph.

The drum is motor driven, and both its position and speed are controlled by means of a stroboscope lamp which is operated by a timer mounted on the engine shaft. This same timer operates a jump spark which furnishes a record of bottom dead-center on the film.

In addition to synchronizing the film drum it is necessary to control the times at which the exposure of the film starts and stops. A shutter located in front of the lens is controlled by an electromagnet which, in turn, is energized by an auxiliary relay operated by the engine shaft. This relay is controlled by a pin which engages with a worm on the engine shaft. The relay contacts close at a definite crank angle, and remain closed for two revolutions of the engine. In this way the magnet that opens the shutter is energized for two revolutions of the engine shaft and holds the camera shutter open for the same length of time.

Results with Cathode-Ray Set-Up

Figs. 13 and 14 show sample oscillograms taken with the improved high-speed apparatus. These diagrams show clearly the benefits of extending the time axis to permit accurate measurements. A difference in behavior is noticeable for the two fuel nozzles. In Fig. 13 where the pintle-type nozzle was used, the entire fuel injection is over before combustion starts whereas in Fig. 14, it may be seen how the fine holes of the four-hole nozzle hold back the fuel and injection lasts over a considerably longer period of time.

Adapted to Many Combustion Studies

This high-speed indicating system, recording as it does three simultaneous impulses, furnishes a means for studying a great array of combustion problems for both Diesel- and Otto-cycle engines. Detonation in the gasoline engine, behavior of fuel dopes, and spectrum analysis in both types of internal-combustion engines can be studied. A microphone can be substituted for one of the other pick-ups to study the sound accompanying fuel knock. Also, light or pressure can be picked up at different parts of the combustion-chamber, and the rate of travel studied.

The first major project was scheduled for the summer of 1936. About 24 Diesel fuels were studied in both the present test engine and in a commercial high-speed unit having a separate combustion chamber. The 24 fuel samples on hand include a great variety of classifications. There are both straight-run and cracked distillates, besides blended residuals. Products of California, Mid-Continent, and Pennsylvania crudes are all represented.

Acknowledgment

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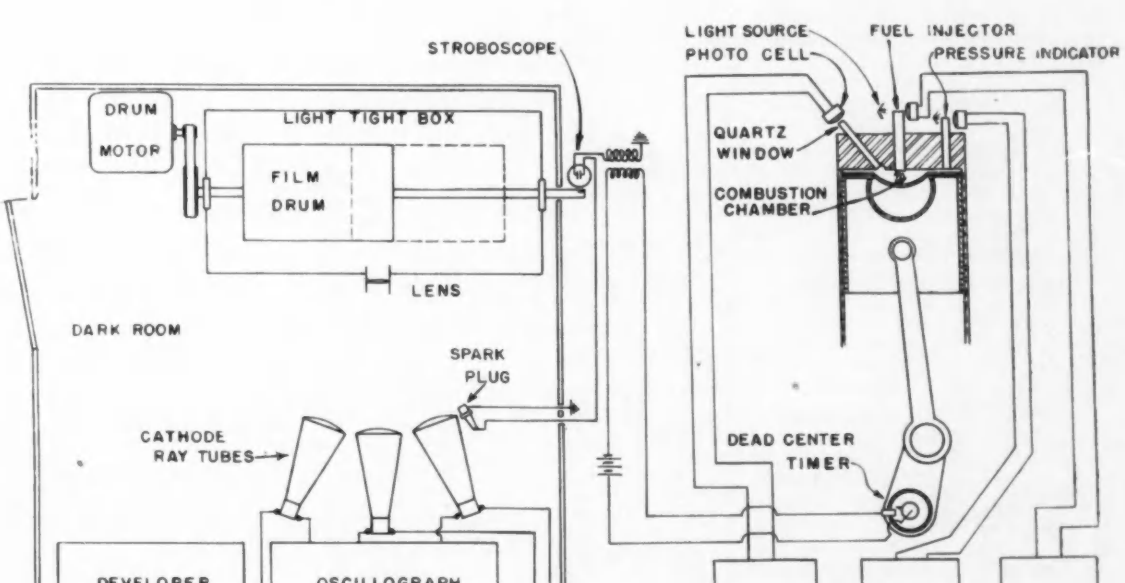

Fig. 12 – Cathode-Ray Combustion-Study Apparatus

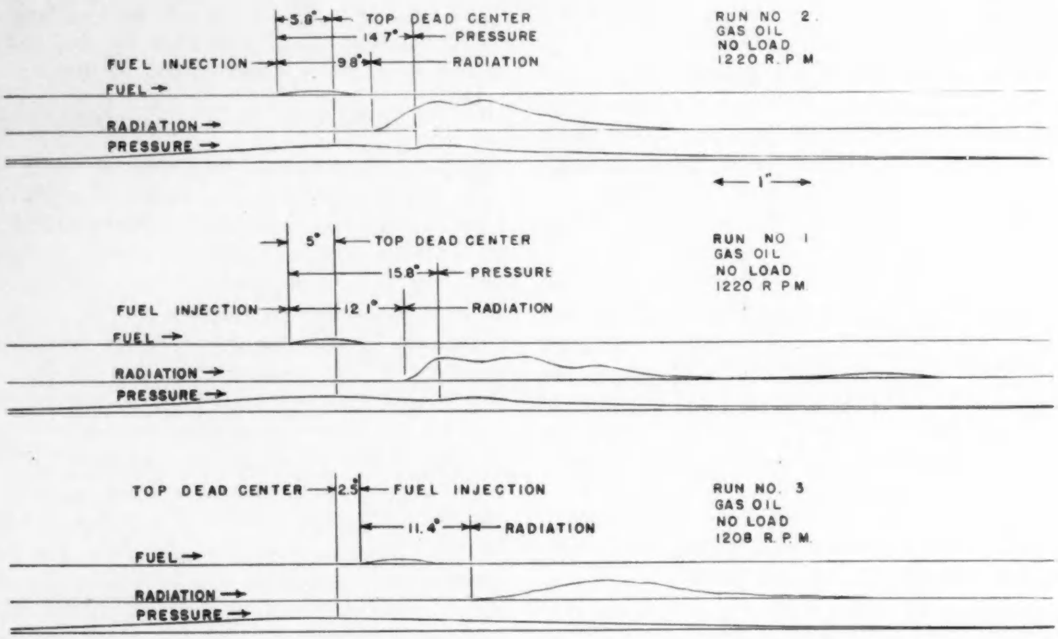

Fig. 13 – Cathode-Ray Oscillograms for Gas Oil and Pintle-Type Nozzle

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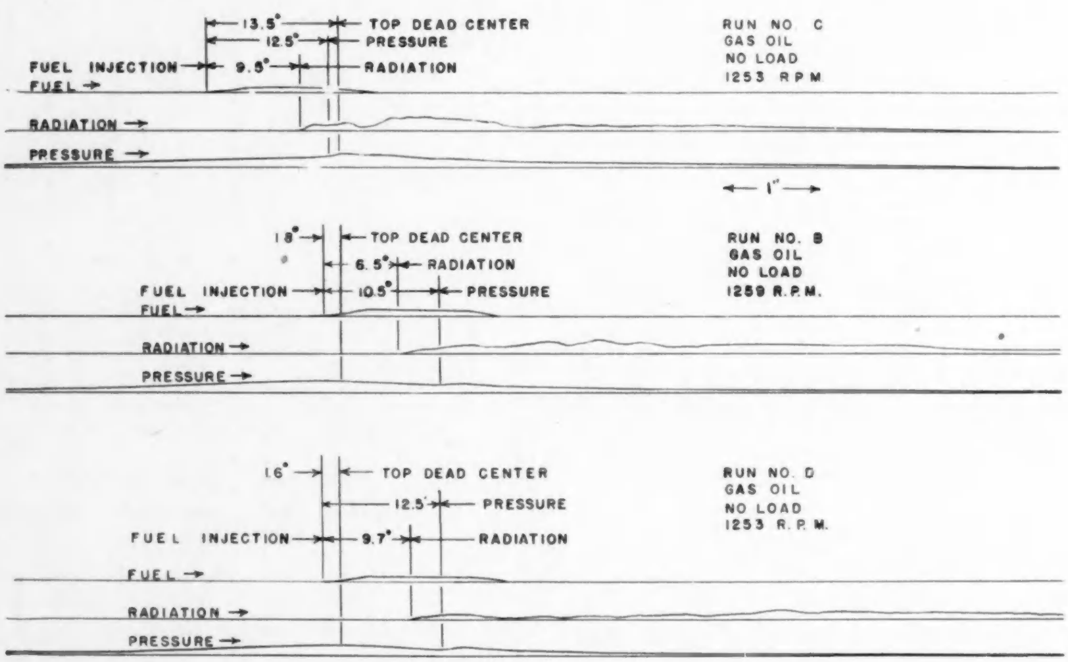

Fig. 14 – Cathode-Ray Oscillograms for Gas Oil and Four-Hole Nozzle